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Helpful Hints for Employers

It is true that there are not enough "live-wire" stationary engineers to fill the demand, and here is the reason:

MOST engineers who study and keep up to date do so because of their love for the work and not for the usually small cash return.

But there are not enough men of this class to meet the demand. The desire to make money is stronger in most humans than love of work.

The necessity for earning money to support and educate a family prevents many "natural-born" engineers from following the vocation of their choice.

First-class power-plant engineering requires brains, knowledge, experience, courage, energy, perseverance, honesty, love of the work, etc.

Call it what you please, but men possessing these gifts are usually alive to the fact that they are valuable, that Providence intends them to use these qualities for their own as well as for the general good.

Therefore they take the gifts for sale to a market where adequate prices are paid.

Employers Take Notice!

Contributed by JAMES E. NOBLE, *Toronto, Ont.*

Welds—The Weaknesses and Merits of Their Structure

By P. A. E. Armstrong*

SYNOPSIS—The author believes that welding is unquestionably the future means of joining the sheets of pressure vessels. The structure of the metal of the plates and that in the weld are entirely different ordinarily, though alike in chemical composition. The high temperature to which the metal being welded is subject is usually of such long duration as to cause serious growth of the crystals. Burned structure is the result of incipient fusion and is caused by the crystal beginning to melt at the crystal boundaries. The plates to be welded should be "tacked" the whole length of the joint, and the weld completed by filling the V for its entire length with thin layers of the filling metal, one layer being completed before beginning the next. This avoids internal stresses in the weld from shrinkage of the cast metal and prevents serious growth of the crystals in the plates.

Welding is of immediate and considerable importance to the power-station engineer and to the power-plant manager. Although the engineer is not actually interested in the manufacture of articles that are to be welded, he has to use pressure vessels, boilers, etc., that are wholly or partly welded. Under these circumstances his interest is perhaps a little more pertinent than that of the manufacturer, as the engineer has to be both responsible for these various tanks and live beside them.

Welding Must Receive Deeper Study

Many questions are raised as to the safety of welding, as to the best system and whether welding should be permitted for high-pressure work. It would be as well to deeply consider the essential points necessary for safe welding and try to find the reason why some welds are not good and others are satisfactory. I am convinced that the subject of welding is not receiving the careful investigation that it deserves, as welding is unquestionably the future means of joining one plate to another.

The art of riveting has received considerable attention, and the science has been reduced to a basis where the engineer can make a definite calculation as to the strength and service of a riveted joint. Why should he not be able to reduce welding to such a state that he can make calculations with a fair assurance that they are accurate?

At the present time there are two widely known systems employed for welding—oxyacetylene (that is, gas welding) and electric welding (that is, welding by means of an electric arc and metal electrode). Both these systems have their respective champions, but there are a number of undesirable features present with either, which could be eliminated if the necessary metallurgical knowledge were applied.

Two steel plates that have been welded are joined by a metal of like, or nearly like, chemical composition, but the structure of these metals is entirely different. The plate was originally cast structure, but has been refined by the operations of mechanical work and heat-treatment. The structure of the added metal, due to welding, is cast metal and cannot be subjected, generally speaking, to mechanical work, although it can be subjected to heat-treatment. Weakness of a welded joint is unquestionably due to three things: (1) Faulty welding; (2) the weakness of the structure of the added metal; (3) thermal disturbance in the vicinity of the weld on the plates being welded.

To remove the cause of faulty welding—by that I mean adjacent areas which, although fused, are not fused together so that there are no interposing slag lines or oxide lines—it is necessary that the added metal and the original metal should both be brought to a molten state so that the new melt will flow on the original metal, which is made molten in such a way that the oxide and other impurities float to the surface of the molten metal and do not remain buried in the molten mass and weaken it. This can be readily accomplished by having sufficient heat to cause a very liquid melt and maintaining this liquid melt for sufficient time to enable the inclosed impurities to rise to the surface.

FIG. 1. PHOTOMICROGRAPH, SHOWING ONE LAYER OF 0.20 PER CENT. CARBON (MILD) STEEL UPON 0.45 PER CENT. CARBON STEEL

There is no sign of crystal growth even immediately adjacent to the added metal which has a "pine tree" structure. The black is pearlite, white is ferrite. The added metal is at the top

The restrictions placed upon this condition are those of increased size of the crystal structure of the plate to be welded, because of the period of high temperature to which it is submitted; therefore it is necessary to use very intense but extremely local heat. In preventing grain growth we are assisted by the original plate, owing to the lag present in the crystal changes, time playing an important part. It is possible to heat a piece of low-carbon steel to a temperature up to fusion without increasing the size of the crystals, but to do this it is necessary that the heat be applied only for a very short period.

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Fig. 1, which is a photomicrograph, shows a piece of steel about 0.45 per cent. carbon, which has had added to it mild steel deposited by means of a metal electrode and the electric arc. This photomicrograph is of great interest, as it shows the little understood and less appreciated fact that the crystal structure of steel is changed by time multiplied by heat and not heat multiplied by time. This is of no moment where ordinary heating is concerned, but where welding is to be considered it is indeed important.

The rapid growth of the grain size of mild steel is very marked if a piece of mild steel plate is heated to, say, 1000 deg. C. (1832 deg. F.) for but a short time. The crystal structure will be found to have grown to such an extent that the grain size is readily discernible to the eye without magnification, and if the steel is still further heated, the crystals take on a coarse angular structure common to overheated steels.

"Burned Metal" Merely Incipient Fusion

The burnt structure, about which so many statements have been made, generally to the effect that in welding the structure of the steel adjacent to the weld must necessarily be burnt and for that reason useless, does not hold true in the instance as shown in Fig. 1. There is no burnt structure here. Burnt structure means incipient fusion and is caused by the crystal beginning to melt in the portions of the crystal which froze last, when it was originally cast. This can occur either at the crystal boundaries or in the heart of the crystal, but is generally at the crystal boundaries. All commercially made steel contains occluded gases—oxygen, hydrogen, nitrogen, etc. Directly incipient fusion starts, the occluded gases, particularly the oxygen, combine with the metal that has started to melt, forming an iron oxide which is readily discernible under the microscope. Such metal is weak along its crystal boundaries, first, because of this iron oxide and, second, because of the "pine tree" structure caused by the solidifying of the molten portions of the crystals which have started to melt at the time of incipient fusion. Such a structure is useless by reason of its great crystal boundary weakness, due not so much to reduced tensile strength as to its poor resistance to shock or alternating stress. All burnt metal is intercrystallinely weak, whereas normal mild steel is strongest to shock load at its crystal boundaries. From this it is readily understood that there are sound metallurgical reasons why the fusing of the metal to be welded should not be done with an agent that is not extremely local and intensely hot.

Gas welding gives good results for thin plate work, because the heat is sufficiently local and intense to make a satisfactory weld in such time that the original plate does not suffer materially from incipient fusion; but where a thicker plate, for instance over $\frac{1}{8}$ in., is concerned, then oxyacetylene sets up distinct thermal disturbance in the vicinity of the weld, due to the prolonged time the flame has to be applied to the part to be welded to raise it to the fusing temperature.

The electric arc with metal wire as an electrode is not much better because of the method of application, although the metal electrode used with the electric arc is an extremely local method of heating.

The work to be welded is generally V'd and the entire V is filled up and sometimes reinforced at one application. That is to say, the weld is started and the V at the cross-section is filled up higher than the level of the surrounding metal as the electrode is brought along the work. In this respect it is practically the same as the oxyacetylene. The electric arc, although being very local and extremely hot, is kept in one position for such a long time that there is distinct crystal growth in the vicinity of the weld and incipient fusion takes place. This system therefore does not have any advantages over the oxyacetylene and in fact is not as good, because the cooling of the metal is quicker and the cast metal is in an extremely chilled state, whereas with oxyacetylene, although the thermal disturbance near the weld is much greater, the added metal is not in such a chilled state and in many instances is found to be stronger and more ductile, this being largely due to the added metal chilling much more slowly.

Fig. 3 shows two plates V'd out and welded in the usual way by the oxyacetylene method. Fig. 4 shows these two plates welded by the electric arc usual method. Fig. 5 shows two plates welded and the V filled up by a number of layers, each running the entire length of the welded plates.

Figs. 3 and 4 are subjected to enormous strains due to the contraction of the cast metal; steel shrinks about $\frac{3}{16}$ in. to the foot. A plate $\frac{1}{2}$ in. thick with a V of 90 deg. will have an area to be filled in at the top of the plate

FIG. 2. SHOWING REFINING EFFECT OF HEAT BY ADDING THE METAL ONE LAYER AT A TIME

This fine structure represents the best structure obtainable in annealed low carbon cast steel. Note the uniformity of structure and absence of segregation

of about $\frac{3}{4}$ in. wide, and when the sides have been melted, probably $\frac{7}{8}$ to 1 in. The molten metal will then have to shrink at the top about $\frac{1}{16}$ in. The major part of this shrinkage takes place when the metal is at its weakest stage; that is, at the time of the solidification and in the plastic stage. In the event of the plate not lifting or pulling sufficiently to take care of this shrinkage, the cast metal of the weld will be found to crack for some distance in the weld. These cracks are more noticeable in electric welds than in gas welds, because in the latter the metal is put down in a number of globules which are spread over by the action of the flame in layer form and the shrinkage is partly taken care of by these layers and partly by the slower freezing of the added metal. The metal electrode and the electric arc deposit larger quantities of metal, the operation being much quicker, but the

shrinkage is not so well taken care of and there is a too rapid chilling.

The benefits derived from gas welding are, however, offset by the added thermal disturbance in the surrounding plate because of the time necessary to take care of this slower application.

Fig. 5 shows the metal put down in a number of longitudinal seams. It is not possible to so weld two pieces of metal with the oxyacetylene torch in its present state of development and show any sort of economy in time and cost. The electric arc alone is applicable to this method, and even then the electrode employed must be of such construction as will lend itself to this type of welding. This is being done, and the welds thus produced have been found to be far superior to welds made according to Figs. 3 and 4. The thermal disturbance in Fig. 5 is extremely small.

Fig. 1 is a photograph of the structure of a piece of steel which has been welded one layer thick by this method. As will be seen, there is no thermal disturbance in the vicinity of the weld, but the added metal shows distinct rapid cooling and must be refined. Each successive layer that is put down has the effect of heat-

thin as possible. The strongest weld by this system is the one where the greatest number of narrow thin layers have been made.

The time employed in making such a weld is of necessity slightly greater than where the system shown in Figs. 3 and 4 is employed, but the additional time does not amount to very much and the increased cost is not sufficient to warrant the rejection of the system on this ground.

The method of welding Fig. 5 also makes it possible to set the plates to be welded so that they are gapped in parallel and not set 1 in. or so apart at one end and touching at the other, as is the case with plates some feet long which are welded as in Figs. 3 and 4.

If two plates of ½-in. steel are to be welded, each plate being 2 ft. long and V'd 90 deg. and welded by oxyacetylene under the normal conditions as per Fig. 3, then one end of the plate will be made to touch and the other end will be set apart from 1 to 2 in., according to the skill of the operator, and it will be found that when the plates have been welded their entire length, the end at which the plates have been set apart is so closed that the edges are touching. This has set up an enormous

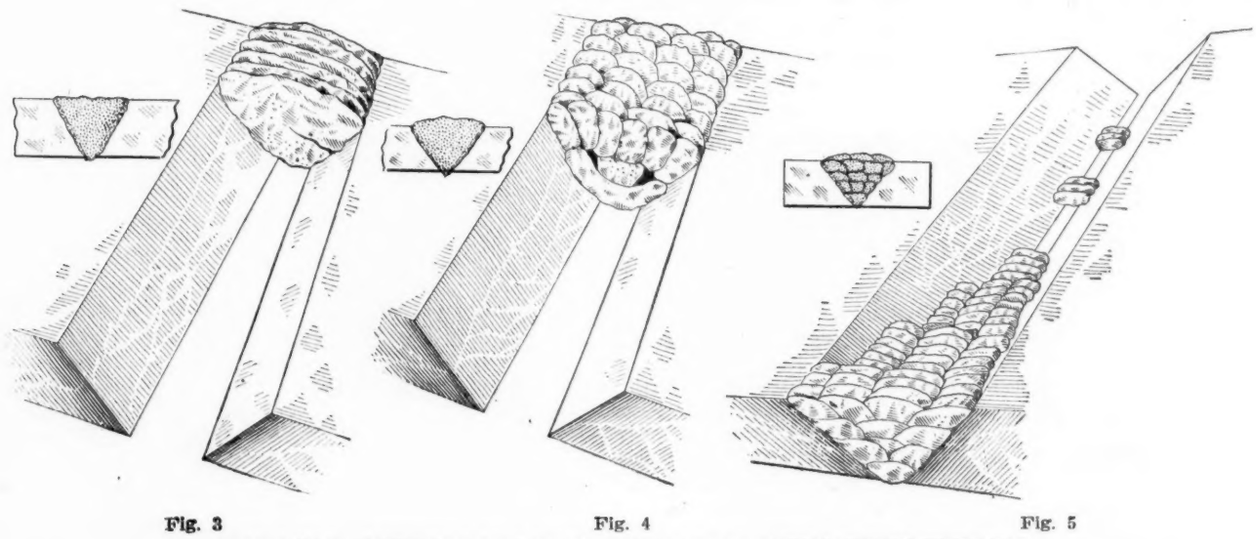


Fig. 3 Fig. 4 Fig. 5

FIGS. 3 TO 5. METHODS OF LAYING DOWN ADDED METAL WHEN WELDING

Fig. 3—Oxyacetylene welding, filling complete in one operation. The wide gap is drawn together by the contraction of the added metal in the weld, tending to break through the weld

Fig. 4—Electric arc metal electrode welding, filling in one operation by carrying electrode from one side to the other. Gap is closed as in Fig. 3

Fig. 5—Electric arc metal electrode welding, filling by putting down one thin layer upon another, thus heating. Treating each layer when laying down the next. The bottom of the V is tacked

treating the under layer, thereby refining the grain and giving a very small, dense structure, as seen in Fig. 2, which was taken in the heart of such a weld.

It will naturally be asked, If this method of welding does not set up crystalline growth in the original steel, owing to the heat being of such a short period, then how can the heat caused by successive layers have any effect upon the structure of the cast metal which is the weld? The answer is this: (1) The heat necessary to cause crystal growth must be much greater than the heat necessary to refine the grain; (2) the cross-section of the layers is extremely small compared with the cross-section of the original metal. Therefore the heat conveyance or radiation of the narrow layers is not sufficiently quick to take away the heat before the crystals have had time to change. The heat set up in laying down a fresh layer of metal is sufficient to refine the under layer, but it is nevertheless necessary that these layers be narrow and as

strain throughout the weld, which undoubtedly materially affects the strength of the plate, particularly as far as resistance to shock is concerned.

If these two plates are welded by means of the metal electrode and electric arc, as per Fig. 4, the plates will not be set so far apart, but nevertheless there will be an enormous strain set up in the weld, because of the constant closing in of the plate. In fact, many such welded plates, when tapped with a hammer so that a rhythmic vibration is set up, will be found to crack and spring apart, perhaps for half their entire length.

Two plates of the same thickness and length welded as per Fig. 5 are not set apart more at one end than at the other, but are set consistently about ⅛ in. apart top and bottom. The plates will then be welded in spots (tacked) about every 6 in. along the length of the plate and afterward welded the entire length one thin layer thick. The plate is now a solid mass—joined the whole

length. The metal is not subjected to any strains, the thin layer having been quite capable of taking care of its own shrinkage, which was extremely small. As each successive layer is put down, the shrinkage is taken care of in the layer. The layer is of much less cross-section than the plate and therefore does not have any tendency to lift the plate and close in the top of the V.

Having shown the mechanical method of overcoming abnormal shrinkage in the cast metal and a way of refining the cast metal by successive layers, it is logical to consider the methods and conditions under which the metal is made molten:

Protection Against Oxides and Gases

All metal that is melted must be protected from the effects of the atmosphere, to prevent oxides forming and the metal becoming saturated with oxygen and nitrogen. The manufacturer of openhearth steel is troubled with oxides and nitrides which permeate the metal due to gas impregnation arising from the partly consumed air in the heating gases. The electric steel furnace is not so troubled, because the melt takes place in a neutral atmosphere, the arc consisting of vaporized lime—in other words, a vaporized slag arc. If it were not for this type of electric arc, and should the arc consist of incandescent atmospheric gases, then the melt in the electric steel furnace would be distinctly inferior to the openhearth melt.

For thin plate work oxyacetylene is a very desirable method of welding from a quantity production point of view provided the flame can be kept neutral. In practice this is not possible, because of the changes in the pressure of the oxygen and acetylene gases. This defect can be remedied, however, if an automatic valve is designed so as to control the mixing of the gases and maintain the flame always in the neutral condition. I do not know of any such arrangement, but there is no reason why this should not be done, and it is one of the desirable developments that should be undertaken by the manufacturers of oxyacetylene welding apparatus.

Welding by oxyacetylene is generally done by means of an oxidizing flame. This cannot be traced on the surface of the steel, but the metal is full of occluded oxygen and therefore extremely weak in its resistance to shock, although in some instances stronger than the more oxygen-free metal as far as tensile strength is concerned. If the flame is carbonizing, which can easily happen, the weld is still troubled with occluded oxygen, because of the affinity of oxygen for carbon. The carbon deposit is partly consumed by the oxygen of the atmosphere, and the excess oxygen impregnates the added metal to the detriment of the weld. Therefore the weld is very porous, being full of occluded gas and iron carbides. If this is carried to excess, the deposited metal is actually white iron, 2 per cent. carbon, in which case the weld cannot be filed and is in an extremely brittle state.

The electric arc, where the metal electrode is used, is entirely an oxidizing one, the arc being composed of incandescent atmospheric gases. The iron is so quickly rusted away by the highly oxidizing nature of the arc that red rust abounds everywhere in the vicinity of the arc and can be found not only in large quantities on the work, but in the nostrils of the operator. As previously pointed out, it is not possible to make good steel while it is being subjected to the oxidizing influence of the atmosphere. Therefore metal deposited by means of the electric arc, wherein no provisions have been made to eliminate the oxygen or degasify the melt, must necessarily give a very weak structure and contain iron oxide, particularly between the added metal and original plate.

Direct Current Sets by Electrolysis

Another difficulty present in the electric-arc welding arises from electrolysis. It is the practice almost without exception to weld with direct current, using the work to be welded as the positive pole and the electrode to be melted as the negative pole. This is done so as to have the hottest part of the arc where the area of metal is greatest. In theory, this is good, but direct current sets up electrolysis and the impurities are collected on the positive side of the line, likewise the oxides. Under these circumstances the weld is literally permeated with impurities by electrolysis. If the poles were reversed, then the metal deposited would be freer from the bad effects of electrolysis. Unfortunately, it would be very difficult to make a good weld with the bare wire electrode having the work as the negative pole. The metal would be deposited all right, but there would be no fusion or only partial fusion between the deposited metal and the original steel. The heat developed would have the effect of melting away the electrode much quicker than the original plates, owing to the larger cross-section and the more rapid heat radiation.

May Coat Electrode with Refractory Tube

The only means of overcoming this difficulty is to localize the effect of the arc in a much more marked manner than can be accomplished with the ordinary metal electrode and electric arc. This can be done by coating the electrode with refractory material, thereby directing the arc inside a tube, as it were, onto the work and getting the full benefit of the heat arising from the electric arc. It will be found that there is very little difference in the heat of either pole. Such refractory materials would melt away quickly enough to maintain a uniform distance between the end of the metal electrode and the work. Furthermore, the atmosphere of the arc would then be that of its nearest gases and, like the electric furnace, would be the atmosphere arising from the gasified refractories. The atmosphere of such an arc would be but slightly oxidizing, if not quite neutral.

With this type of electrode it is possible to use alternating current which is quite free from electrolysis, thereby enabling the steel to be deposited free from the harmful effects of electrolysis, or if direct current were used, then the work would always be the negative pole.

In testing welds it is important that they be tested in such a manner as to anticipate failure through fatigue. The cast structure, which is the welded material, is fairly strong as far as tensile strength is concerned. Its elastic limit is quite high and very near to its tensile strength; the elongation is very low, likewise the reduction of area. To submit welded pieces to the usual tests of tension, elastic limit, elongation and reduction of area does not throw as much light upon the strength of the weld as is desirable. If a piece of 0.35 per cent. carbon steel is welded, the tensile strength of the welded metal is perhaps 10,000 lb. under the tensile strength of the original plate and the elastic limit of the welded metal may be only 200 lb. under its tensile strength. In pulling the sample the original plate will start to "neck" away from the weld gener-

ally, if an oxyacetylene weld, about ⅜ in. to ½ in. away from the weld; that is, in the area that has been annealed, or overheated, owing to the thermal disturbance. This necking sets up reduced cross-section; the weld not having suffered from reduced area, is now of greater cross-section than the original steel. The break does not occur in the weld, but in the original steel.

It is erroneously thought that the weld is as strong as the original plate, but this is obviously not so. In the case of an electric weld by means of the ordinary bare wire, the elastic limit and tensile strength of the added metal of the weld are about the same, and the tensile strength is probably two-thirds the tensile strength of a 0.35 per cent. carbon plate. Necking will take place away from the weld, likewise the break, and the weld is erroneously thought to be as strong as the original steel.

To test a welded plate, it is necessary to first test an unwelded piece of the same steel and then prepare the sample so that the welded area is necked, the necked portion not being taken back as far as the edges of the weld. The sample then being pulled, the true tensile strength, elastic limit, elongation and reduction of area of the welded metal will be discovered.

Advises Hydrostatic Hammer Blow Test

In all cast metal, if the elastic limit is high, the metal is dangerous, because it will be found to have very little so-called anti-fatigue value. It is important to test this value where pressure tanks are concerned or where vibrations are likely to occur. Impure cast metal—that is, metal high in impurities, phosphorus and sulphur—or metals rich in combined gases are all liable to sudden rupture when submitted to shock load. Therefore it is important when testing welds, that a shock or vibration load test be submitted. I would advise that a sample piece that has been welded should be submitted to an alternating-stress test machine, or even tested by a rotary-stress test machine, of the Arnold type. The finished tank, on which tensile-strength tests cannot be made, should be submitted to water pressure, there being no danger in such testing, and the welded joint tested by a number of blows by the hammer or by a specially designed machine that will deliver a given blow at a given speed, traveling at a predetermined rate. A weld thus tested, if faulty, will immediately break, and a good weld will not be damaged by such treatment. There is a very wide margin between good and bad cast steels which have been broken during shock and vibratory tests. Steels that are faulty will break quickly, whereas steels that are capable of resisting shock load will stand quite a lot of shock testing before they break.

In alternating-stress testing, the crystals of the metal begin to show signs of slip along their cleavage planes, and unless these cleavage planes actually show signs of fissures, the metal has not been seriously damaged. In fact, a polished specimen of very mild steel can be tested to, say, 30,000 reversals at 15-ton load and the amount of deformation set up along the cleavage planes will not have reached the danger limit, and the signs of slip along these cleavage planes can be repolished out and cannot then be traced under the microscope.

When testing for resistance to alternating stress or shock, weak metal will quickly break because of crystalline weakness, whereas the normal break of good steel is directly across the cleavage planes and not around the crystal boundaries, as such boundaries are considerably stronger mechanically than the body of the crystal.

Testing a welded pressure tank and boiler by rhythmic hammer blows is not impact-shock testing but alternate stressing, due to vibration, and the hammer blow is never sufficient to set up deformation or cold working of a serious nature. There cannot thus be any objection on the score of premature damage to an otherwise sound tank by this method of testing.

Equipment for Testing Motors

By H. L. Hearvey

In order to test alternating-current motors, transformers, etc., repair shops have found it necessary to have at hand alternating current of a wide variation in frequencies, voltages and phases. The testing plants of the different establishments range from a very elaborate outfit to a single motor-generator or converter to take the available direct current and convert it into an alternating current of one frequency and voltage.

A converter operated from the direct-current side produces a very flexible unit when the power is taken from a three-wire 110- and 220-volt system. One type of converter which is built especially for this purpose gives 40 cycles when operating normally on 220 volts. By inserting a rheostat in the shunt-field circuit, as shown in Fig. 1, the armature speed can be increased, making it possible to amplify the frequency to 60 cycles. In this manner a range of from 40 to 60 cycles is obtained without any change in the set except the field current. If a lower frequency is desired, the double-throw switch *S* is thrown to the up position, putting the armature across

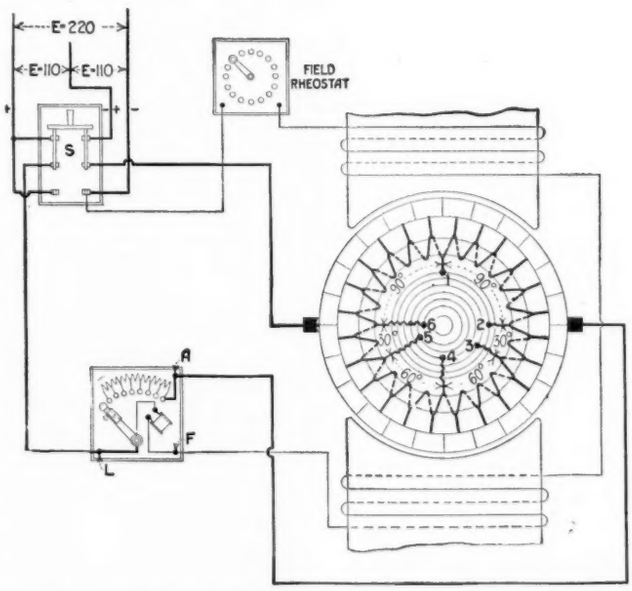


FIG. 1. DIRECT-CURRENT CONNECTIONS TO CONVERTER

the 110-volt lines and 220 volts on the field circuit. This combination gives 20 cycles. Again, by the use of the field rheostat the frequency can be raised to 35 cycles.

By the use of six slip rings connected to the armature, as shown in Fig. 1, one, two and three phases can be had. Single-phase current is obtained from 1 and 4, and also from 2 and 6. These two sets when used together

give a four-wire two-phase circuit. From rings 1, 3 and 5 a three-phase current is obtained, making it possible to test one-, two- and three-phase equipment from the same source.

The single-, two- and three-phase voltages from the converter with 120 volts at the commutator will be approximately 85, 85 and 73 volts respectively. With 240 volts at the commutator the voltage at the collector rings for the different phases will be double these values. These voltages may be corrected by two transformers having a voltage ratio of 1 to 1.7 with a tap taken out of the high-voltage winding to give a ratio of 1 to 1.5. By the use of two three-pole double-throw switches connected as shown in Fig. 2, the transformers may be connected to give approximately 120- or 240-volt one-, two- or three-phase current, depending upon the pressure at the commutator.

With both switches closed toward the transformers, 120- or 240-volt three-phase current can be obtained on the secondaries, depending upon the voltage at the commutator. Throwing the switches away from the transformer connects the primary of the latter to collector rings 1 and 4 and 2 and 6, which are the single-phase and two-phase connections. The secondary switch is now connected to the 1.5 section of the winding, which corrects the primary volts to approximately 120 or 240, depending upon the direct-current voltage. By extending the secondary winding and bringing out proper place taps, most any combination of voltage may be obtained to meet the requirements.

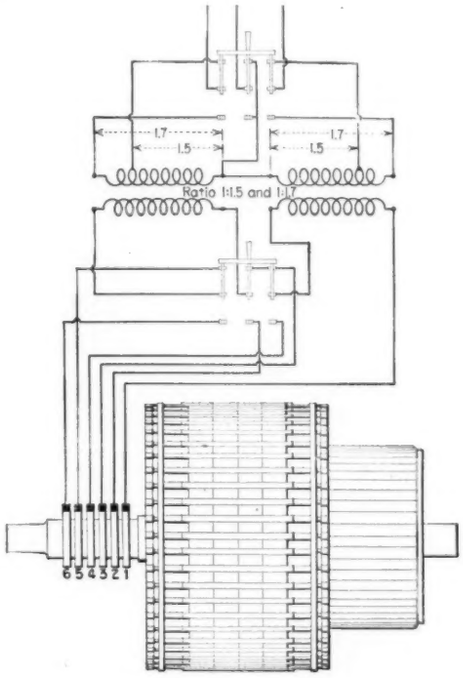


FIG. 2. TRANSFORMER CONNECTIONS TO CONVERTER

The question of starting current should be considered when a testing plant is being laid out. The starting current required by many of the small alternating-current motors, especially of the single-phase variety, is often three or four times the full-load current. Since the torque of a motor varies as the square of the applied voltage, it is evident that there should be no marked decrease in the converter's voltage if the torque of the induction motion under test is not to be seriously impaired.

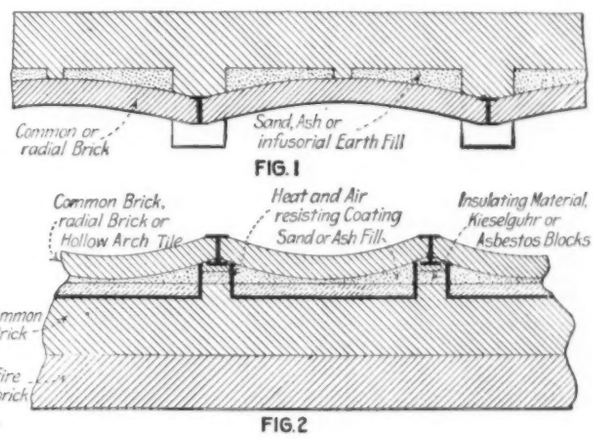
Efficient Boiler Settings

By Joseph Goder*

In the Apr. 17 issue of *Power*, page 535, there is a letter relating to cracks in boiler settings, but no means of remedying the conditions are suggested. Apparently the blame for cracks is placed upon the faulty suspension of the boiler. It is the writer's opinion that the manner of suspending the boiler is of less importance than the proper construction of the brickwork. The method of suspending the boiler from gallows frames, if well and properly done, serves satisfactorily in all cases. The fault lies in the construction of the masonry. Buckstays and binder rods, well arranged, will help to a great extent, but will not overcome defective setting.

A change from the customary straight wall setting to the construction shown in Fig. 1 assures a practically "crack-proof" setting. This type is employed extensively in Europe, especially in Germany, where its use is almost universal. The higher cost will be well repaid by the resulting increase in efficiency.

Instead of building a straight wall, the external course is arched and the arches are braced by I-beams, which also serve as buckstays. The inner course is firebrick, laid as usual, bound and backed by an 8-in. common brick wall


Common or radial Brick
Sand, Ash or infusorial Earth Fill
FIG. 1
Common Brick, radial Brick or Hollow Arch Tile
Heat and Air resisting Coating
Sand or Ash Fill
Insulating Material, Kieselguhr or Asbestos Blocks
Common Brick
Fire brick
FIG. 2

FIGS. 1 AND 2. BOILER SIDE-WALL CONSTRUCTION
Fig. 1—External course arched to permit expansion. Fig. 2—Improved side wall for high-duty boilers

transmitting its outward-acting stresses to the crest of the arch. The space between the straight wall and the arches is packed with sand, ash or infusorial earth filler.

A modification of this setting, Fig. 2, answers the most rigorous requirements demanded of a wall for high-duty boilers. The backing of the straight wall with a heat-resisting plastic coating reduces the infiltration of air. Insulating material reduces the radiation losses to a minimum. The elasticity of the brick arches provides the necessary expansion allowance. A boiler setting, well constructed, of this design is at least equal, if not superior, to a steel-incased straight setting.

The most troublesome part of the setting of a return-tubular boiler is the rear arch. Means must be provided to take care of the expansion of the brickwork as well as of the movement of the boiler shell. A construction that behaves very well under the most severe conditions is shown in Fig. 3.

*Boiler efficiency engineer, Chicago.

Short tee-irons rest with one end on an angle fastened to the boiler and the other end free to move on the brickwork. These tees carry the refractory material. The

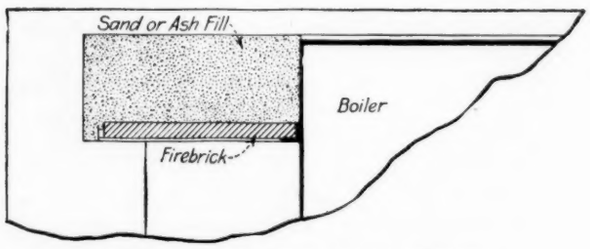


FIG. 3. GOOD REAR ARCH DESIGN

steel may be protected against the heat by fireproof cement. A layer of clay, ashes or kieselguhr is placed upon the top of the tiles and topped with a course of brick or even cement.

The covering of the top of the boiler drum may also be improved by the construction shown in Fig. 4. If a return pass over the top of the boiler shell is desired, the construction shown in Fig. 5 may be used. The rests for the top arches serve, in this instance, as tie rods for the I-beams.

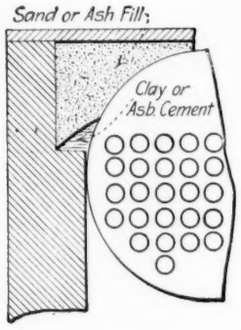


FIG. 4. COVERING FOR BOILER TOP

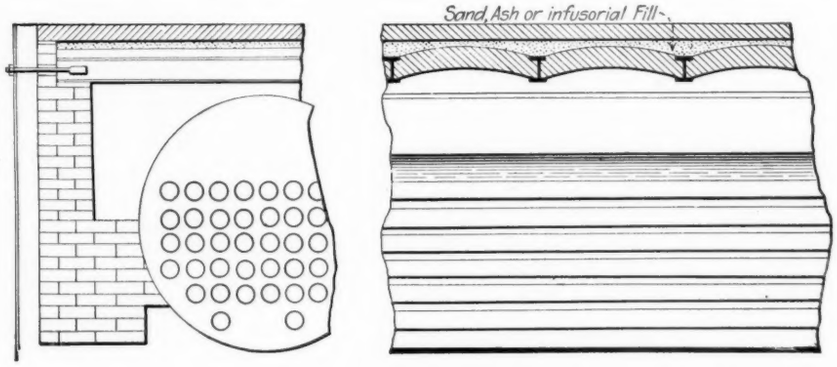


FIG. 5. CONSTRUCTION WITH RETURN PASS OVER BOILER SHELL

The workmanship on the brickwork is of the utmost importance. The employment of experienced men on this kind of work tends to accomplish better results than merely a special design or construction. Too often such important work is turned over to incompetent men because the item of first cost, unfortunately, receives too much attention at the expense of maintained economy. The results can be seen in seventy out of every one hundred boiler settings.

A Simple High Vacuum Pump, recently developed at the United States Bureau of Standards, can be built entirely of glass by a reasonably skillful glass blower and requires for its operation only a bunsen burner and a water aspirator. The remarkable feature of this exhauster is that it is capable of giving vacua down to better than 0.001 mm. pressure.—Commerce Reports.

Large Electric Locomotive—The Pennsylvania R.R. Co. has recently completed an electric locomotive designed to operate on 11,000-volt single-phase 25-cycle current taken from an overhead trolley. The primary of a static transformer is connected between the trolley and the rail, with the secondary connected to a phase converter, which changes the single-phase current into three-phase for use in the traction motors. These motors, of which there are four, are rated at 1200 hp. each, giving the locomotive a capacity of 4800 hp. Water rheostats are used to control the motors, which, when connected in parallel, will run the locomotive at a speed of 20.6 miles per hour. The motors can also be connected in cascade to give a speed of 10.3 miles per hour.

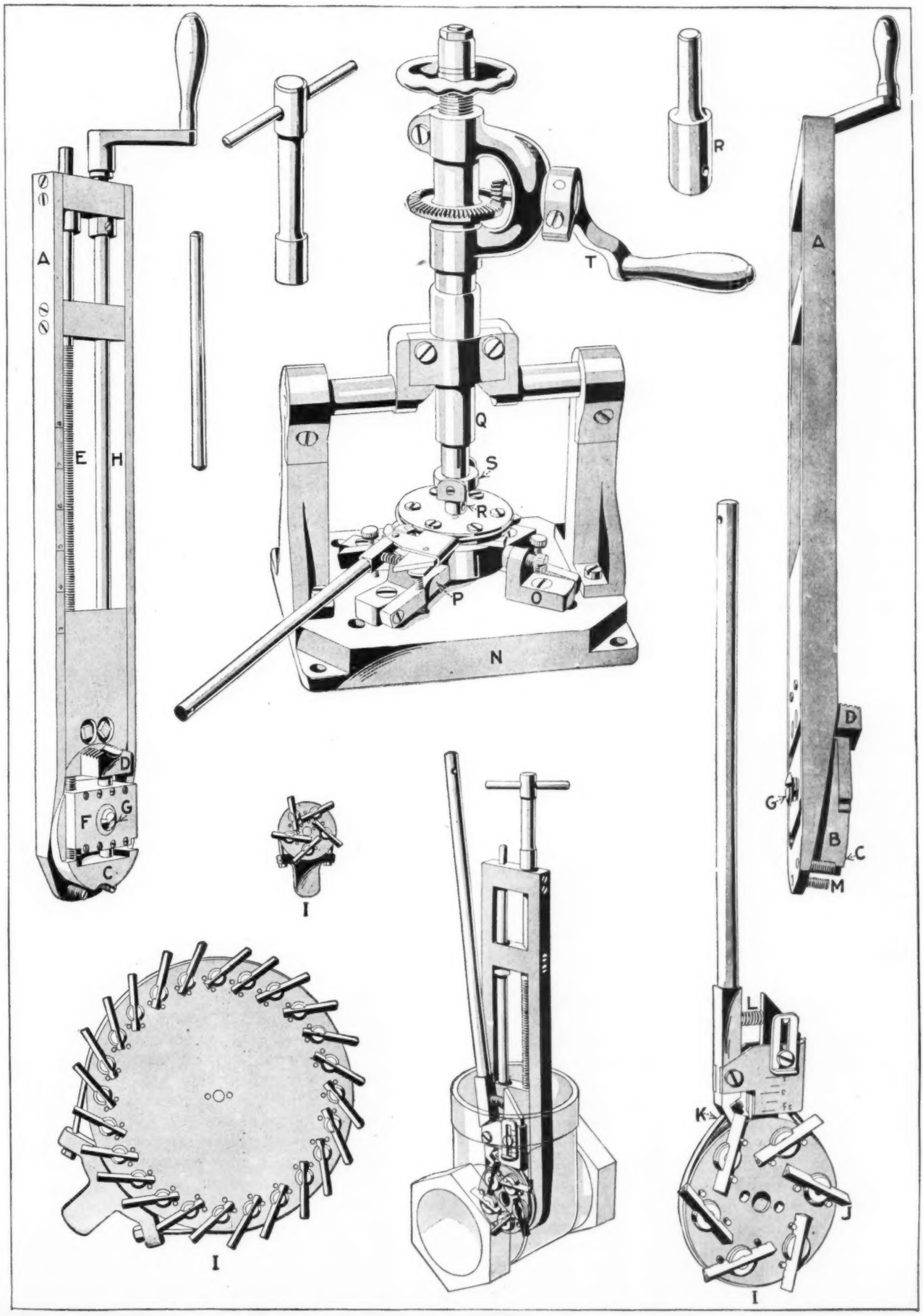
Dexter Gate-Valve Reseating Machine

It is an easy matter to regrind the seats of globe valves and this, together with the renewable disks, has enabled the steam user to keep this type of valve tight against the moderate pressures with which it is used and to greatly prolong the useful life of the valve.

With gate valves working with high steam pressures and superheat, difficulty is experienced in keeping them tight. It is a difficult and tedious job to remove one from a pipe line and chuck it properly in a lathe for refacing the seat. With a renewable seat there is the possibility that the ring will be distorted and leak because of the expansion of the valve body when steam is turned on. A remedy would be to regrind the seat ring if a suitable machine were obtainable.

Machines for regrinding gate-valve seats and disks have been constructed by the Leavitt Machine Co., Orange, Mass., but they had the disadvantage of not being capable of recutting the seating surface of both the straight and angle types of valve seats and preserve the four surfaces of the disk and seat at the same angle. The company has, however, just put on the market a new device known as the Dexter Gate-Valve Reseating Machine. It is capable of reseating all designs of gate valves from 1½-in. to 14-in. size, without disconnecting the valve body from the pipe line. The machine is designed to recut the seat and gate, also split disks without changing the angles in relation to each other, thus making it possible to take any valve with a badly worn seat ring and gate and make it tight. The recutting can be performed as many times as the thickness of the seat will permit, when a new set of rings can be put in and the life of the valve prolonged indefinitely. A set of new seat rings can likewise be made tight, should the distortion of the valve body spring them.

This tool for recutting gate-valve seats consists of two members, a cutter support and a cutter head. The cutter support *A* (see illustration) is in the shape of a frame, the bottom end of which is beveled at *B* to conform to the beveled type of seat. A stationary jaw *C* is at the bottom end of the support, and an adjustable jaw *D* is moved up and down by a threaded spindle *E* that projects above the support head for a crank attachment. The lower end of the spindle is made with a thread of one-half the pitch of that which moves the movable jaw and moves a centering head *F*, containing a cutter-bearing stud *G*. As the movable jaw is to grip the inside



DETAILS OF GATE VALVE RESEATING MACHINE AND CUTTING HEAD

of the valve orifice, the centering head is moved one-half as fast in either direction as the movable jaw, so that when the cutter support is secured in place, the centering-head spindle is central to the valve seat. A second spindle *H* operates a set of spiral gears that rotate an internally threaded sleeve, which carries the cutter-bearing spindle and forces the cutter head against the seat.

The cutting head *I* consists of a disk with from five to twenty V-shaped cutting blades *J* (according to the size), each one having six cutting edges, so that when one edge becomes dull it can be turned and the other five edges used in succession. Two of the edges are for roughing work. The edge of the disk is made with a groove in which a friction band *K* is fitted. A spring *L* between the handle and anchor plate keeps the friction band clamped against the cutter head. Pressure exerted against the spring releases the clamp, and when the handle is pulled in the opposite direction the friction of the band is increased and any further movement in that direction causes the cutter head to turn. This construction provides a movement similar to that of a ratchet, but it is superior as there is no appreciable wear.

Recutting the Seat Ring

Recutting the seat rings is a simple procedure. The first thing is to select a cutter-bearing stud of suitable length (the stud is removable), so that the cutter head can be held firmly against the seat to be refaced. The cutter support is then placed in the valve, resting the lower jaw of the support on the inner circle of the valve-seat ring opposite the one to be refaced. These supports should be placed at about the same angle as the seat, whether it is perpendicular or at an angle. When seats on an angle are to be recut, the two back-stop pins *M* (used with large supports) are removed.

Having clamped the jaws of the support in the valve by rotating the threaded spindle, select the proper-sized cutter head and adjust the cutter blades so that the outer end of each will cover the outside circle of the valve seat, using the gage on the cutter head to obtain the proper adjustment. The cutter is then placed in the valve and the center bearing extended from the head by turning the spindle until the bearing head comes in contact with the center of the cutter head, thus holding it to the seat. By rotating the cutter with the friction lever, at the same time feeding the centering pin slowly toward the cutter head, the seat is refaced. If the seat is badly worn, the roughing or sawtooth edge of one side of the cutting blades is first used until a fair surface is obtained. Then the cutters are turned and the smoothing edges employed to make the finished surface.

Refacing the valve wedges, either solid or split, is accomplished by placing the wedge on a bedplate *N* and holding it in place by four dogs *O* that permit of centering the disk with the tool spindle. All dogs are adjustable on the bedplate and are bolted in place. A disk can be leveled by turning screw standards on the dogs, the disk being supported by the levers *P*.

When the disk has been centered and secured by the dogs, a cutter similar to that used for recutting the valve seat is placed on the disk and the spindle center *Q* is screwed down until the oval end comes in contact with the center of the cutter head. The cutter is then rotated with the friction lever while feeding slowly by the hand-wheel at the top of the spindle.

For removing scale and gloss on the seating surface, the sawtooth cutter is first used and then changed to the finishing edge. To obtain the polished surface after roughing and refacing, remove the pin *R* and screw the centering spindle down with the pins in the universal jointed head *S'* engaging in corresponding holes in the cutter head. Holding the cutter handle and turning the detachable handle, the spindle and cutter are revolved by means of the beveled gears. By lightly feeding the centering spindle, a smooth polished surface of the disk is obtained.

As the expense of removing a large valve and putting in a new one is often nearly equal to the cost of the valve itself, it is apparent that the saving of a few valves within the range of the tool will quickly pay for the cost of the outfit.

Wattmeter in Wrong Location

By A. T. Walker

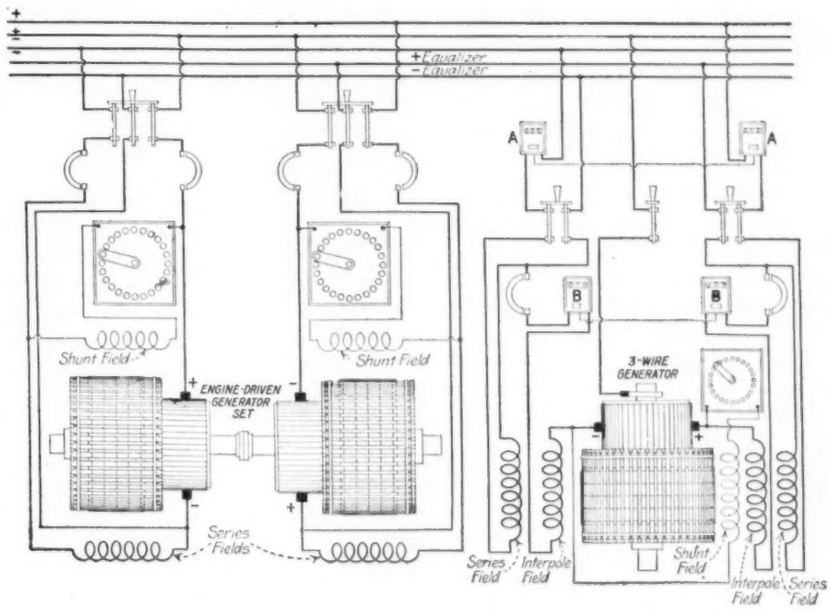
In an isolated plant where central-station service had been introduced to help carry the load, through a motor-generator set, a mistake made in the location of the wattmeters by the installers caused considerable confusion before the error was discovered. The central-station equipment consisted of a three-wire motor-generator set, a maximum-demand meter in connection with two watthour meters, etc. The purpose of the demand meter was to automatically record the number of kilowatts supplied by the machine during every half-hour period.

The generator set supplied direct current at 120 and 240 volts to the three-wire system for lighting and power. The engineers were instructed to keep a close watch on the indicating wattmeter and ammeters and not to allow the recently installed machine to supply more than a predetermined amount of current to the busses. Whenever it became evident that the load requirements exceeded this predetermined number of kilowatts, an engine-driven generator was to be started and put in parallel with the motor-generator or the load reduced by pulling some of the power switches.

There was at first no attempt made to check up the ammeter readings with those of the maximum-demand meter, the assumption being that the central-station erectors had connected up the apparatus properly; and no reason for doubting this showed up until the first maximum-demand meter readings were taken and the first bill for current rendered. Then it was found that the maximum demand had considerably exceeded the point above which it was not to be allowed to go. The engineers were at first held to account apparently for not having watched the line ammeters and starting an engine or pulling some switches at the proper time. In order to keep a constant check on the amount of current used, the engineers were thereafter required to read the figures registered every 30 min. on the maximum-demand record slip and compute the current consumption hour by hour.

Not long after this system was put into effect, it was discovered that there was a discrepancy between the amount of current going through the motor-generator-set ammeters and the amount as registered by the maximum-demand meter. As long as the motor-generator set was the only machine in service, the readings checked up all

right, but when an engine-driven generator was put in parallel with the former, the ammeters and demand meter did not always check. Also, with the engine generator in parallel with the motor generator, if the breaker on the latter was opened, the wattmeter on that set would sometimes keep on recording until one of the main switches was pulled. All this indicated that there was something radically wrong, and an investigation was started which developed the following: The contract with the service company stipulated that the watthour meters should be placed between the generator and busbars; this was done so that the consumer would not be charged up with the current required to operate the generator set. The figure shows the main connections as they were first installed, with the wattmeters at *A*. Note that the wattmeters are between the main switches and the busbars. Tracing these connections will show that with the motor-generator-set breakers open and the main switches closed, current from the engine-driven generators can flow from the equalizer busbars through the series fields and thus act on the wattmeter coils. The same condition would prevail if the breakers were in and the voltage of the motor-generator set low, so that the engine set carried more than its share of the load. Under these conditions current from the engine-driven set would be going through the motor-generator wattmeter and demand meter and still not be indicated on the motor-generator-set ammeters which were located between the circuit breakers and the interpole field winding. The greater the difference between the engine-set and motor-generator-set voltages, the larger would be the discrepancy between the amount of current supplied by the generator set and the amount recorded on the maximum-demand and wattmeters.



CONNECTION DIAGRAM OF THE TWO-WIRE AND THREE-WIRE GENERATORS

The fact that such a discrepancy existed soon led to the reason; the wattmeters were then placed at *B* in the figure, between the breaker and interpole winding.

The Selection of a Lubricating Oil

By H. J. Macintire*

SYNOPSIS—Briefly describes the accepted theory of lubrication of bearings. Tells of the fluid pressure maintained in a journal and how this is affected by speed and weight of journal and viscosity of the oil. Various tests for lubricants are described and illustrated.

Given two samples of lubricating oil, which is the better? The answer cannot be given offhand as those acquainted with the subject know. It is not similar to the selection of gaskets or packing, but more like the purchase of coal. The problem of lubrication involves considerable theory modified by the kind of lubrication, the design of the bearing, the kind of service, the speed, unit pressure, and other lesser considerations. Each of these factors must be considered as well as the effect of their combinations. To understand the problem at all it is desirable to study the subject from the standpoint of the heating of the bearings, the action of the lubricant as regards the formation of the oil film, viscosity, adhesiveness, fluid friction, etc.

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Consider the weight *P*, Fig. 1, sliding on a horizontal surface with uniform velocity. A force *F* is required to maintain this speed. As there is no change of speed involving an acceleration, we can say that the resistance to motion equals the driving force, similar to the condition with traction vehicles moving on a level road. For small velocities this may be said to be due to friction only, and by definition we may put

$$Pu = F \text{ or } u = \frac{F}{P} = \textit{coefficient of friction}$$

If this force *F* is applied for a certain time (a minute) during which the distance *S* is traversed, the work done is

$$W = F \times S = Pu \times S$$

where *S* is the distance in feet per minute. The heating effect *Q* will be

$$Q = \frac{W}{778} = \frac{Pu \times S}{778} B.t.u.$$

Assume a shaft of weight *W* rotating in a journal of diameter *d* and at a certain speed of *N* revolutions per minute. The heating effect per minute will be

$$Q = \frac{Wu\pi dN}{778} B.t.u.$$

where 778 is the mechanical equivalent of heat. As an example, if the weight is 2000 lb., the journal 6 in. diameter, the speed 100 r.p.m., and the coefficient = 0.1, then

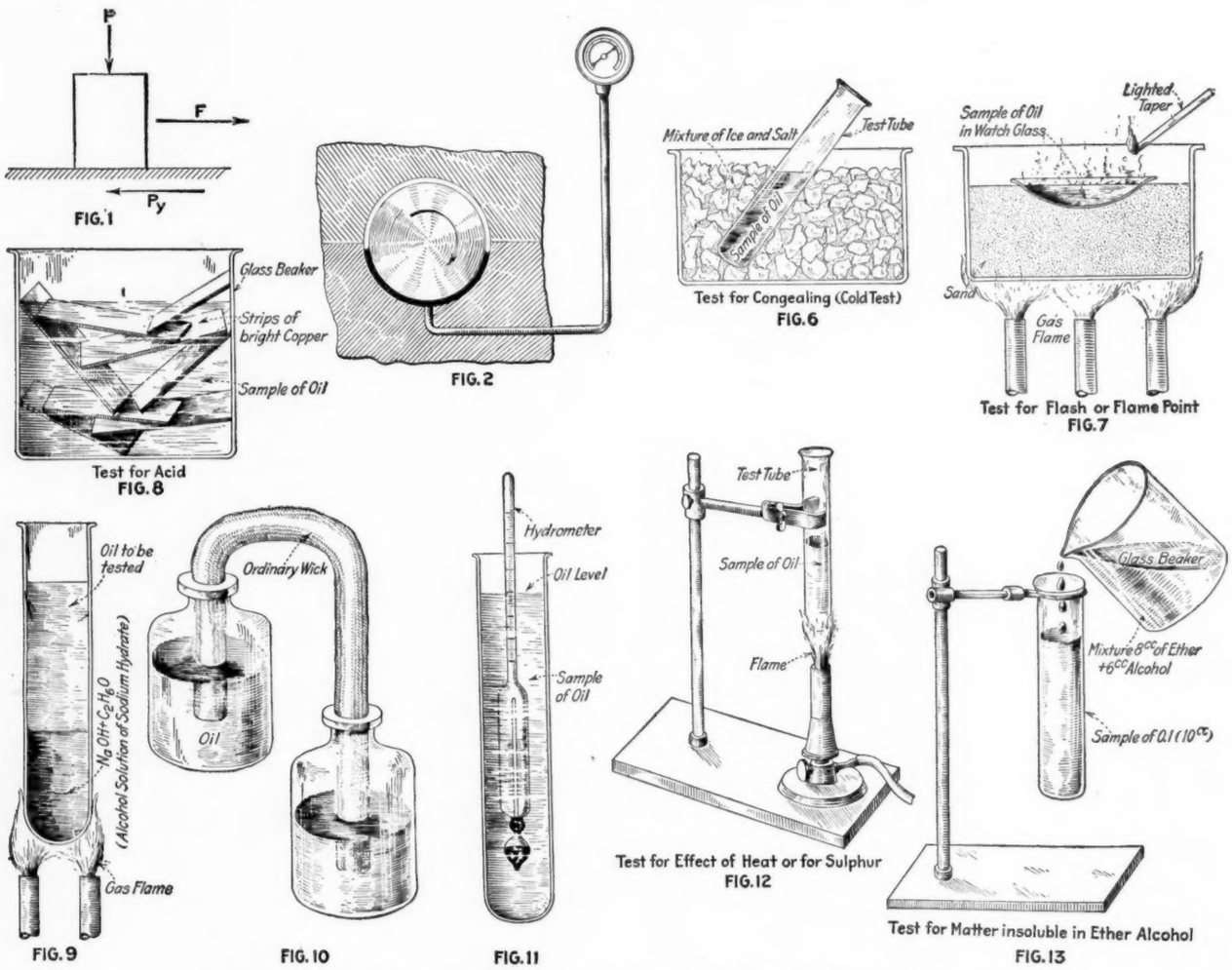
$$Q = \frac{2000 \times 0.1 \times 3.1416 \times \frac{1}{2} \times 100}{778} = 40.38 \text{ B.t.u.}$$

per minute.

This loss of power and its resultant heating due to friction cannot be countenanced in power transmission. It is clear that the heat generated per unit of time must be dissipated during the same time or the temperature of the journal will increase. In this case serious trouble might arise owing to melting or seizing of the bearings. It is also seen that too large a percentage of power (10 per cent. in the case cited) is lost beyond recall. From the equations for friction loss, the only way around this difficulty is to decrease the value of u (the coefficient of friction) and thereby cause a decrease in the loss of power and the heating effect—an end achieved by lubrication.

According to our theory of lubrication, the liquid actually separates the two solids and metal-to-metal contact no longer exists; but the moving mass floats on a film of oil, and resistance to motion is that due to the objection to the flow of liquids (that is, fluid friction), which may be $\frac{1}{10}$ that due to metal friction or less. Besides a reduction in power loss, greater unit pressure may be used as well as higher surface speeds. As everything in this theory of lubrication depends on the formation of an oil film, it is worth while to consider this product for a moment.

There are a number of distinct proofs of the formation of this oil film. The oldest of these is that used in Tower's experiments. In his investigations on journals, he drilled small holes at various points in the bearing and attached a pressure gage. He thereby obtained a record of the range of pressure exerted by the lubricant in different parts of the journal, which pressure went up as high as 650 lb. per sq.in. That this is a fluid pressure built up and maintained by the action of the oil and the rotating shaft is evidenced by the connections shown in Fig. 2. The second method, that used by Professor Moore, is electrical resistance. While at rest, only metal contact exists. As the speed increases, the adhesive property of the oil enables it to wedge itself between the surfaces and to form a film. By suitable electrical connections, the resistance to the passage of a current can be measured. Evidently this will increase as separation becomes more perfect, and the speed can be determined at which the perfect oil film is developed for any condition of unit pressure and design of bearing. Professor Moore found



FIGS. 1 AND 2 AND 6 TO 13. SHOWING VARIOUS TESTS OF LUBRICATING OIL

Fig. 1—Demonstration of friction
Fig. 2—Showing pressure in journal
Fig. 6—Cold test
Fig. 7—Flash test
Fig. 8—Test for acid
Fig. 9—Test for animal and vegetable oils and mineral soaps
Fig. 10—Test for gumming of the oil
Fig. 11—Specific-gravity determination
Fig. 12—Heat and sulphur tests
Fig. 13—Test for asphalts

that for mineral oils the oil film was formed when the velocity was that given approximately by $v = \frac{p^2}{55}$, where v = feet per minute and p = pressure in pounds per square inch.

In lubrication, evidently, the oil must be fed in at the point of zero pressure and advantage taken of the inherent property of the lubricant so that the relative motion provides a pumping action. This forces the oil under the journal and builds up the pressure in consequence, thereby providing the necessary hydraulic support for the journal. Finally, the oil squeezes out and escapes to the oil reservoir, carrying with it a large portion of the heat generated by friction. The journal then acts similarly to a pump, which is the principle involved in a celebrated reduction gear and dynamometer. In this machine the oil pressure for operating the oil pistons is obtained by pipe connection to the lower part of the bearings. It would seem that there cannot be the slightest doubt that perfect oil lubrication substitutes fluid for metal friction. In such a case it has been proved that the oil in contact with the metal does not move relatively to the metal—that there is no metal-to-oil resistance. Therefore, having made use of the lubricant to decrease friction and prevent abrasion of the bearing, the next step is to select judiciously the particular lubricant for the conditions of operation.

All fluids have the physical property of viscosity, or "tensile strength," but to a varying degree. If one puts a finger into cylinder oil and another into sperm oil, the amount adhering to the finger when it is removed from the oil will be different, depending on the relative abilities to flow or to adhere. If placed in a bearing, the oil's ability to cohere is very important, for the friction developed now depends on fluid friction. That lubricant which is most desirable is the one having the least cohesion (or viscosity) consistent with the conditions of the problem. One would not consider the use of cylinder oil for the bearings of a sewing machine or similar work, neither would light sperm oil be used for work involving heavy unit pressures. Likewise the property of adhesiveness is important. Water may be wiped off the surfaces of solids much more easily than oil. In lubrication this attraction of oil for the surface of the journal is utilized to create and maintain the oil film, which would be very imperfectly developed if mercury were employed. Given then, the problem of determining which is the better of two oils for journal lubrication, let us consider a few preliminary points before making our final answer.

It is customary to use the same oil for successive work, to collect the oil after using it, to filter and to separate as far as is possible any water collected with it. Unless care is taken, the filter will gradually remove the heavier portion (the body) of the oil and minute foreign material will slowly contaminate it. [There are those who do not admit that filtering or using an oil over and over again tends to wear it out. Others contend that the medium through which the oil is filtered determines whether or not the lubricating qualities will be impaired.—Editor.] This decrease of viscosity and specific gravity (See Fig. 5) and also the increase in impurity tend to cause trouble because of lack of ability to maintain the oil film or because of increased coefficient of friction. Conditions of operation vary, the unit pressure may be 25 to 350 lb. per sq.in. of projected area, the working temperature may be at 60 deg. F. or lower or may be in contact with the boiler steam. The surface speed of the journal may be a few or a hundred feet per second (the latter in the case of a steam turbine). The bearing may be carefully made and fitted correctly for just the proper formation of the oil film, or it may not be. If we are to decide intelligently, we must consider the conditions of operation and select the oil for that combination only. There is no one oil "par excellence." If one grade of oil is to be used for several kinds of service, a sacrifice must be made in favor of the case involving the most severe conditions.

Again, oil may be of various origins—the so-called mineral oil may be of asphalt, paraffin or the olefine base, each of which has different physical properties, the viscosity increasing in the order of the list mentioned. The asphalt-base oil decomposes at the higher operating temperatures and thereby ceases to act as a lubricant. Be-

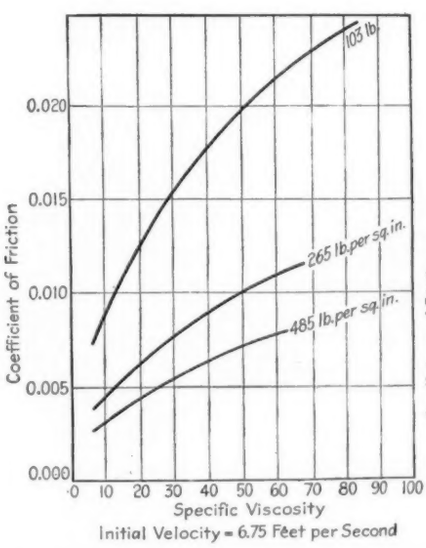


FIG. 3. SHOWING THAT COEFFICIENT OF FRICTION IS FUNCTION OF PRESSURE, VELOCITY AND SPECIFIC VISCOSITY

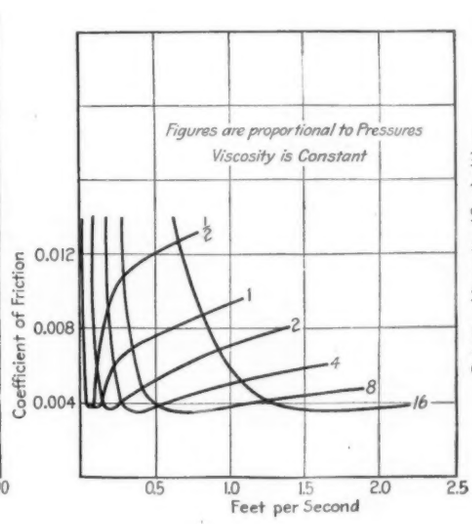


FIG. 4. RELATION OF COEFFICIENT OF FRICTION TO PRESSURE AND VELOCITY

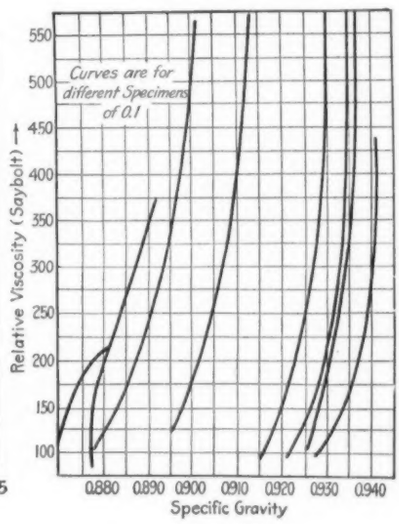


FIG. 5. SHOWING CHANGE OF SPECIFIC GRAVITY WITH CHANGE OF VISCOSITY

sides the mineral, there are the animal and the vegetable oils. These latter do not lend themselves to an increase of temperature—even the change of engine-room temperature may cause the oil film to break down, owing to the decrease of viscosity with increase of temperature, and a hot bearing or worse is the result. It is common practice to test oils for relative viscosity by measuring the relative time it takes the oils and the same amount of water to pass through a small orifice. This method of test is satisfactory for daily engine-room log reports (the specific gravity may be used also—see Fig. 5) and will satisfactorily show any change of quality of the oil and will indicate when it is desirable to add a new supply, yet for purposes of investigation of an unknown lubricant, this is as sensible as to use the temperature of a gas in degrees Fahrenheit involving thermodynamic calculations, instead of the absolute temperature measured from the so-called zero of temperature. We should use the absolute viscosity (or the specific viscosity, which is proportional thereto) for all scientific comparisons.

To resume the previous question, the two samples of oil may be artificially colored; they may be mixtures of various kinds of oil, or the oil may be adulterated with soap, lime, chalk, resin oil, free acid, etc. The engineer should be capable of applying simple tests for these, as well as that for viscosity, at the operating temperature of the oil, and also the cold and flash tests. Acid due to decomposition of animal or vegetable fats, or poor refining of mineral oils, will corrode the bearings. The cold and flash tests are important if extremes of operating temperature are to be met with. The test for gumming is to insure that no decided changes will occur during use on account of resinified products which will increase the friction. These tests must be made if the oil is unknown to the engineer; but by far the most important test is that for viscosity—a fact only recently proved.

Ubbelode on Viscosity

The value of viscosity of an oil was shown by tests and deductions by Ubbelode, which were very extensive and were conducted with oils from America, Galicia and Russia—distilled, crude and refined—some cheap and some expensive. Yet as far as the coefficient of friction was concerned, he could find no advantage in one over another. On the other hand, he found that the coefficient of friction is a function of the pressure, velocity and the specific viscosity. If the latter was known, the coefficient of friction could be calculated. See Fig. 3. There have appeared on the market viscosimeters which will give instantaneous readings of viscosities at any desired temperature. Ubbelode likewise showed the relation of the coefficient of friction to pressure and velocity (see Fig. 4), or that at a certain velocity and a certain pressure the friction coefficient is a minimum.

As mentioned, the engineer should be capable of making certain simple tests for the physical properties and adulterants of the oil, and the following are suggested:

Cold Test—Usually it is sufficient to congeal the oil with ice water or a mixture of ice and salt and then allow the oil to heat until it will flow. The temperature reading should then be taken. See Fig. 6.

Flash Test—Place a watch-glass with the sample of oil in a pan of sand and heat the sand slowly, keeping the oil well stirred. Test occasionally by bringing a lighted taper to within one-half inch of the surface of the oil. The flash point is that temperature at which the vapors ignite with a flash. See Fig. 7. The burning point, likewise, may be found when the vapors passing off the oil will ignite and continue burning.

Test for Acid—The test for acid may be made by adding a little distilled water, shaking and testing with litmus paper. Acid is present when blue litmus turns red. Or the oil may be placed in a glass beaker with bright copper strips and heated slightly. Acid is present when the copper is tarnished. See Fig. 8.

Test for Gumming—Allow the oil to siphon out of a small bottle into another one by the use of an ordinary wick. Make three successive runs of six hours each, and measure the amounts each time. These should be the same. See Fig. 10. The wick allows the air to come in close contact with each particle of the oil and accelerates the gumming action.

Test for Animal or Vegetable Oils or Mineral Soaps—The objection of these oils is due to the formation of fatty acids. The only objection to the presence of mineral soaps (such as oleate of aluminum) is a *seeming* increase in viscosity without increasing the lubricating value. The test for fatty acids is to add an alcoholic solution of sodium or potassium hydrate and heat to 200 deg. F. The soap formed will settle out. See Fig. 9. The test for mineral soaps is to incinerate the oil and examine the quantity of ash. Pure oil should have but a trace of ash.

Test for Specific Gravity—This is a test only for the amount of "body" in the oil. See Fig. 11. It helps to detect adulteration and is a good daily test to make for the condition of filtered oil. Specific gravity may be found from readings in degrees Baumé by the formula,

$$\text{specific gravity} = \frac{140}{130 + \text{Baumé reading}}$$

Test for Asphalt—Asphalts will cause scoring of the journals and clogging of the oil line. The test is to mix ether alcohol with the sample of oil, shake thoroughly and allow the same to stand for 12 hours. Any precipitate that settles at the bottom of the test tube will be asphalt. See Fig. 13.

Test for the Effect of Heat—Heat in a test tube until vapors pass off, and then compare with color of the unheated oil. Sulphur will be indicated by a considerable darkening of color.

SAMPLE SPECIFICATIONS

	Appearance	Specific Gravity	Flash Point, Deg. F.	Flame Temp., Deg. F.	Freeze Temp., Deg. F.	Remarks
Cylinder Oil	Clear reddish-brown or green	0.90	More than 500	More than 600	Less than 60	.....
Engine oil	Light straw	0.90	325	400	5	.....
Dynamo oil	Yellow or light straw	0.87	330	420	5	Not more than 1-10 % mineral acid or 1½% fatty acid.

If it is proved by analysis that the oil is free from adulterants, that it is satisfactory as regards acid, cold and flash tests, etc., then the final decision may be made from the viscosity test only. It will have to be remembered that fast peripheral speeds require less viscosity than a similar bearing under slow speed. Heavy unit pressures require more viscous oils, otherwise there will not be enough body to support the journal. Tight-fitting bearings require less viscosity than loose ones, because the oil is not so readily squeezed lengthwise out of the bearings. Much of this may be seen from Fig. 4, which shows how

the viscosity must be increased for increasing fluid pressures.

In conclusion, one may say that there is usually just one kind of oil that best suits one set of conditions. Since the object of using the oil is to decrease frictional losses, care must be taken not to use too light an oil, as this is dangerous; or too heavy an oil, as this would offer too great resistance and use more power than necessary or desirable. Mixtures of mineral and fatty oils are not advisable because of the unstable character of the latter, which may form fatty acids and corrode the surrounding surfaces. Oil for continued use must be free from oxidating tendencies or gumming, while adulterants to give the effect of greater specific gravity or viscosity are of no advantage. Viscosity tests should be made at the temperature at which the oil is to operate. As mentioned, no oil will serve all conditions, but if it is to be used for several kinds of journals, it must be chosen for the extreme conditions. Finally, if the preliminary tests are satisfactory, that oil comparing best in viscosity to the oil previously in service will be the one to use. If the brand of oil in use gave trouble on account of hot bearings, the engineer should select one with greater viscosity.

It appears, then, that there is no definite answer to the problem stated at the beginning of the article, but that the engineer must find his solution by proper tests and judicious selection. The same degree of care must be taken in the testing and the analysis of the test as is done in the case of flue-gas analysis. The use of an unknown oil on any but an experimental engine is greatly to be deplored.

Continuous Removal of Soot

In the removal of soot from the heating surfaces of boilers and economizers the shorter the interval between the time of the deposit upon the surfaces and the time of removal the easier the removal is effected and the greater the amount of the deposit that can be removed without actually resorting to mechanical scraping.

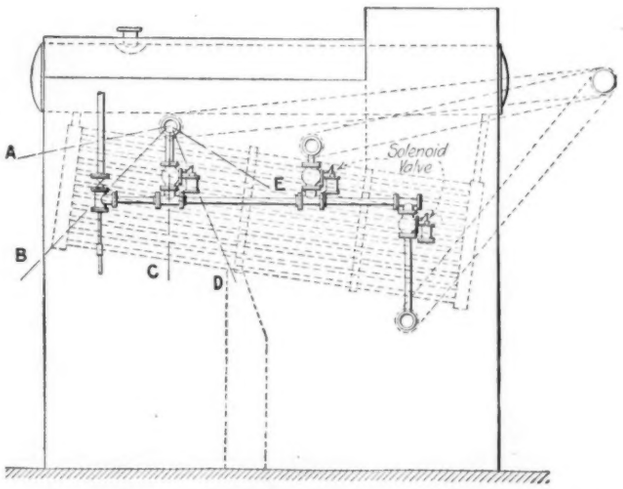


FIG. 1. CONTINUOUS BLOWING SYSTEM APPLIED TO ROTATING NOZZLES

From the viewpoint of thermal conductivity, the shorter the interval between the time of the deposit upon the surfaces and the time of removal the higher the efficiency. A study of the conduction of heat from a gas to water beyond a metal plate discloses the fact that there is a film of very high heat resistance on each side of the metal surface, the resistance on the side of the gases being much higher than the resistance of the film on the side of the water and the resistance of each being many times the resistance of the plate itself.

It is, therefore, not surprising that a small accumulation of soot on heating surfaces greatly increases the resistance to the flow of heat to the surfaces, since each superimposed particle between the gas and the metal introduces not only the resistance of the particle itself to the flow of the heat, but also the resistance of an additional gaseous film.

This accounts for the occurrence familiar to testing engineers where immediately after blowing the surfaces of a boiler or economizer the temperature of the stack gases drops 50 to 100 deg.; and again in many instances this temperature in the course of two hours' time rises to the original maximum previous to the blowing operation.

In pursuance of these ideas the apparatus herein described has been designed for the purpose of carrying out a continuous process of soot removal.

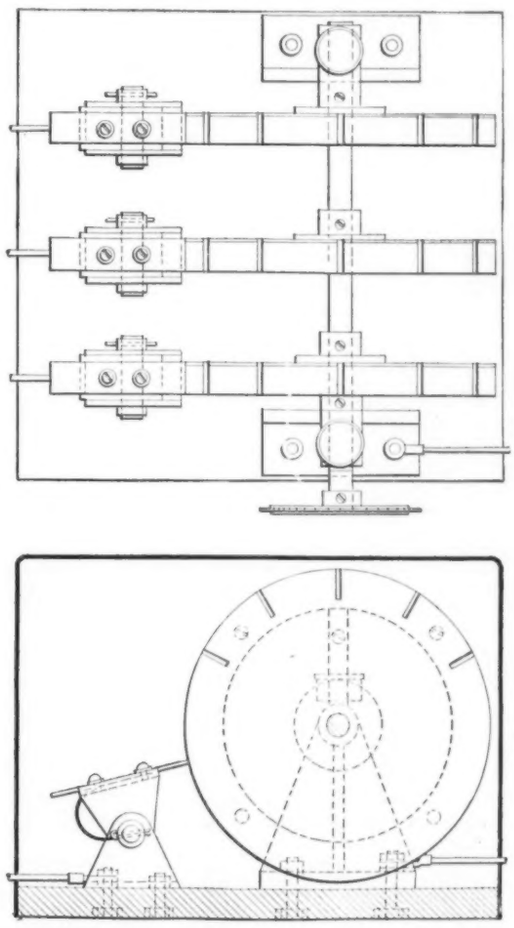


FIG. 2. DETAILS OF CONTACT DEVICE

The process can be applied to practically any of the well-known soot-blowing systems, but is preferably used on those types which comprise stationary nozzles blowing parallel with the surfaces to be cleaned. It can, however, be applied to the rotating-nozzle systems, although the initial cost is considerably greater. Briefly, the pro-

cess is carried out by the insertion of automatically operated quick-opening valves in each branch of the system.

The valves may be either of the solenoid-operated type operated with a contact device similar to an electric sign flasher, or they may be mechanically operated valves actuated by reach rods and a motor-driven camshaft.

These valves, instead of being operated two or three times a day for intervals of five or six minutes, are designed to operate for a very short period of time at regular intervals of several minutes. If each valve is held open for approximately one second and if it is opened every five minutes during the day, it will open 288 times during the 24 hours, or a total period of 4.8 min. This is probably less than the time a hand-operated valve would be open in the intermittent system of operation. The length of these intervals will depend largely upon the nature of the soot to be removed and will vary anywhere from 3 to 15 min. duration, depending upon local conditions.

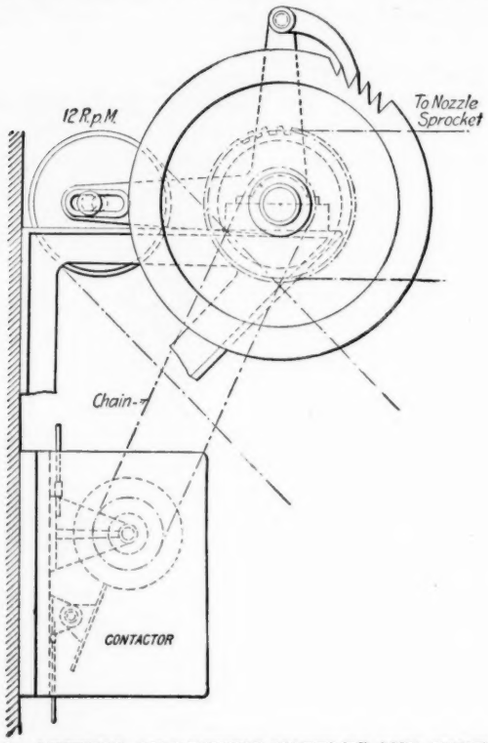


FIG. 3. METHOD OF DRIVING NOZZLES AND CONTACTS

The valves also would be operated in a given order so that in the cycle of openings the valves farthest removed from the source of draft opens first while the other valves follow in the order of their removal. This prevents the soot removed from the first pass of water-tube boilers having an opportunity to settle on the heating surfaces in the second and third passes that have already been cleaned.

Fig. 1 illustrates the application of the system to a rotating-nozzle type of soot blower. A solenoid valve is located in each branch leading to the nozzles. These valves are operated through the contact device shown in Fig. 2. This device consists of three disks mounted on a shaft and carrying contact points. Five are shown, and each contact point corresponds to a blowing position in Fig. 1, as shown at *A, B, C, D* and *E*. There is a contact disk for each solenoid valve.

The rotating nozzles are chain-and-sprocket driven by a lineshaft, the sprockets being located one on each nozzle and one for each nozzle on the driving shaft. Fig. 3 shows the method of driving the nozzles and the contact device from a lineshaft A ratchet arrangement is used for driving the lineshaft, as the speed of this shaft is very low.

Fig. 4 shows a detail of the bearing supporting the rotating nozzle. This bearing is much larger than is common, as it is necessary to rotate the nozzle continuously and with the smallest amount of energy.

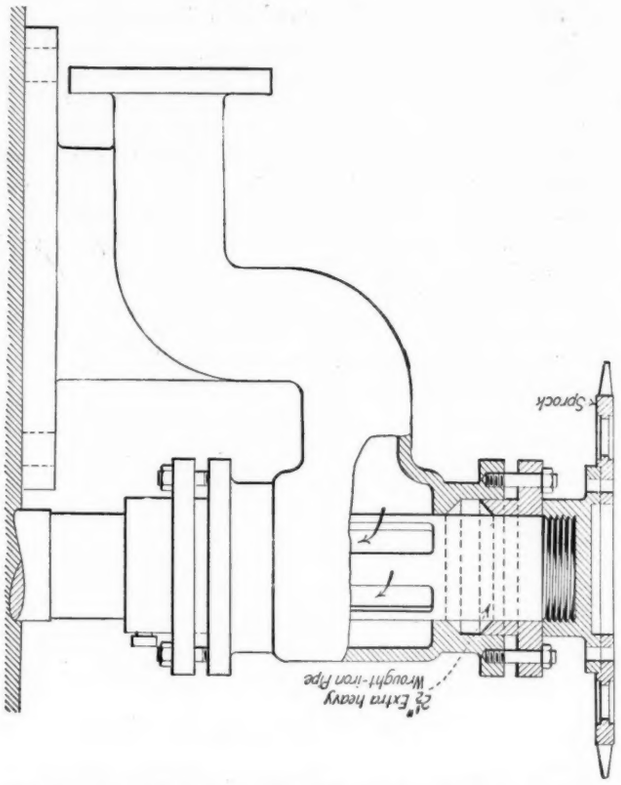
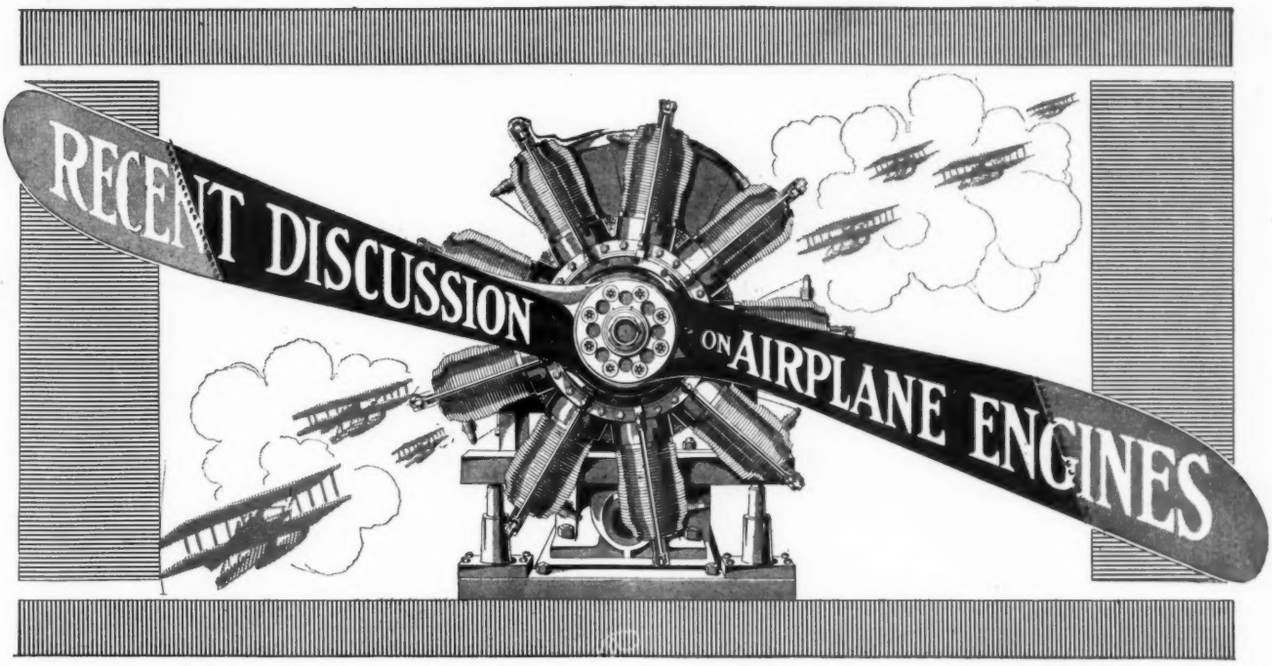


FIG. 4. BEARING SUPPORTING THE ROTATING NOZZLE

When this system is applied to the intrabaffle type of soot blower or other stationary-nozzle type, the solenoid valves are inserted in the branches of the system and the contact device is driven through a small motor or clock mechanism.

This system has been perfected by C. M. Garland, 1049 Ardmore Avenue, Chicago, Ill.

Power Plant at Anchorage, Alaska—Work has been started by the Alaskan Engineering Commission on the construction of a power plant at Anchorage, to furnish the light and power in that place and to the buildings, docks and railroad terminal of the Government's road in the north. The plant will use coal from the Matanuska coal field now being opened up by the Federal railroad. The boiler plant will consist of four water-tube boilers, set in two batteries of two each. The capacity of each boiler is about 175 hp., or a total of 700 hp., when working at a pressure of 150 lb. per sq.in., without superheat. An open type of feed-water heater will be provided, with a capacity for heating 15,000 lb. of water per hour from a temperature of 50 deg. to 205 deg. F. Exhaust steam will be utilized for heating the machine shops by a specially designed heater. Two direct-current generators of 75-kw. capacity each, directly connected to tandem-compound high-speed engines, will furnish power to the shop for driving the machine tools and traveling crane. The first unit for providing energy for light and power outside the shop consists of a 100-kw. 2300-volt 60-cycle three-phase generator, driven at 3600 r.p.m., by a Curtis steam turbine. A second unit of three times this capacity will be installed next spring. The distribution in the yard and town will be 2300-volt three-phase three-wire system, with single-phase three-wire secondaries. All energy sold will be metered.



SYNOPSIS—Excerpts from papers by a lieutenant of the British Royal Flying Corps, Charles M. Manly of the Curtiss Aëroplane Co., Dr. Charles E. Lucke of Columbia, Leigh M. Griffith and others. The engine with stationary cylinders is used where greatest reliability is demanded and rotary engines for fast scout machines. Development of the ignition system, particularly the spark plug, is urged. The water-cooled engine is more reliable and lighter per horsepower, fuel consumption considered, than the air-cooled engine. Dr. Lucke advocates an engine with base of steel trusses, steel tubes for the cylinders, jackets, etc.

The very extensive use of the airplane in this war, together with its severe service conditions, has brought forth some valuable discussion relative to the design of these machines. Officers of the British Royal Flying Corps, now visiting the United States, state that since the war the builders of engines with stationary cylinders have improved their product so that in the airplanes now in use there are about as many of this type as there are rotary engines.

The Speed Is Considerably Over 100 Miles Per Hour

The rotary is lighter per horsepower and is used largely for the scouts and some of the faster battleplanes. It is further claimed that the climbing power of a rotary engine is greater when used in a tractor than that of the engine with stationary cylinders. In the large battleplane reconaissance machines, where the weight is great and where there must be the most reliable power, the stationary water-cooled type is used. They are also used on bombing planes.

The following is from a paper by Charles M. Manly, chief engineer of the Curtiss Aëroplane Co., read before the spring meeting of the Society of Automobile Engineers:

"The most important advances that are to be expected in the next two years are in the improvement of aviation engines. We have some good engines now, but the engines of two years hence will probably be immensely superior to those we now have, especially in the matter of weight per horsepower and in dependableness. The thing requiring the greatest development in order properly to increase the dependableness of the engine is ignition. This includes the magneto, the circuit-breaker and the spark plug. The one thing that is needed most just now is a spark plug that is durable and dependable and used in an engine with a high mean effective pressure. The more important points of the spark plug that require great improvement are air-tightness, electric insulating qualities, resistance to the intense heat, combined with requisite mechanical strength and freedom from causing preignition troubles so as to insure the most dependable and rapid ignition of the whole charge."

There follow excerpts from a paper by Professor Charles E. Lucke, of Columbia University, read at the spring meeting of the American Society of Mechanical Engineers at Cincinnati:

The Water-Cooled Motor Superior

"The air-cooled motor has entirely failed in comparison with the water-cooled motor. The reasons are perfectly sound. The two-stroke cycle engine has given way to the four-stroke cycle type. Fixed cylinders have prevailed over rotating cylinders for most types of planes in the service.

"Taking up each of the factors of airplane design in a specific way, it seems proper to first consider the value of efficiency and the relation of efficiency to minimum weight. Plotting hours of running as abscissas against weight of engine with fuel and oil as ordinates for the air-cooled and the water-cooled types of motor respectively, so that the intercept on the vertical axis represents the weight of engine metal alone and the ordinates away from the axis represent the weight of metal plus fuel and oil, one finds that the two curves cross at some period of running beyond which, therefore, the water-cooled heavier engine, because of its low-

er fuel and oil consumption, becomes lighter in comparison. The metal weight of the water-cooled motor is about 1½ times that of the air-cooled motor. The consumption of the air-cooled motor is approximately twice that of the water-cooled motor.

"Of conditions for efficiency which bear upon this question of fuel weight and which have led to the selection of the water-cooled motor, the first is compression. The higher the compression the higher the efficiency, and there is no limit until preignition occurs. The amount of compression possible is limited by the metal temperature and by the temperature of the mixture as admitted. The next factor is the quality of the mixture. Mixture proportion must be accurately controlled, more accurately than is possible with any existing carburetor.

"It can be shown that the explosion line of the indicator diagram following compression must be maintained vertical for maximum efficiency. The rate of propagation is the one factor that tends to hold it vertical; that is, flame propagation with reference to the piston speed. If the flame-propagation rate is high enough for a given specific speed so that the explosion line is vertical, the efficiency will be high. But should the piston speed exceed a certain value, the explosion line will begin to lean toward the expansion line, with a consequent large loss of work area and low efficiency or high fuel consumption. There is therefore a limited piston speed for every given mixture that cannot be exceeded without destroying the efficiency, and we are now approaching that speed in airplane engines.

Change the Character of Ignition

"The falling off of mean effective pressure at high speed may not be due primarily to volumetric efficiency but to other causes, and recognition of this stimulates a search for those causes. The first is too slow combustion or too high piston speed. This is to be corrected by an additional ignition source, or by moving the spark plug from the side wall to the center point, igniting at more than one point or at a more central point, will cure this defect.

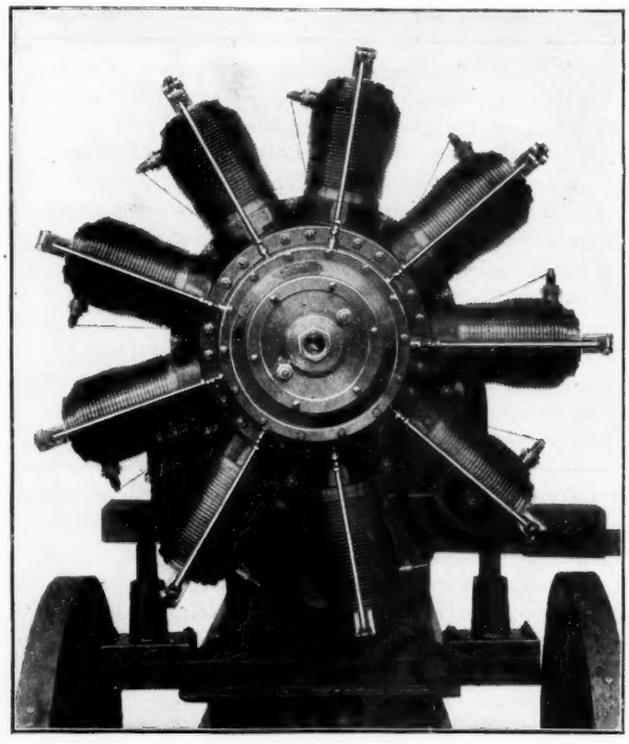
"It is curious that most people have played with cams and adjusted them back and forward by guess and have never bothered about the air meter, which is the only positive means of arriving at best cam forms and valve timing for sustained mean effective pressure at high speed."

Commenting on cylinders Professor Lucke said that one favorable and improving step had been cast iron for the cylinders, head and head jacket in a one-piece casting with sheet metal over the cylindrical barrel for the water jacket. Proceeding along the same line of weight reduction, the next step is to cut away the cast iron joining the ends of the ports and forming the wall of the head jacket, and substitute sheet metal welded to the ports by the oxygen welding system. Wherever there are connections to be made for attachment of gears, there must be some additional supports welded or brazed on.

There is a fundamental objection to cast-iron cylinders for aëronautical work, and it is a valid one. Cast-iron cylinders do not have to be very thick to be amply strong so far as the gas-pressure stresses are concerned, but the fact remains that so long as they are cast iron no one knows whether they are of good cast iron inside or not, and cast iron cut down to ⅛ in. in thickness incurs taking some chances. Here attention is turned toward steel. Drawn steel or forged steel is a reliable material and a logical selection.

Dr. Lucke here described several methods of making cylinders from drawn steel. He brought out the fact that the steel cylinder is naturally flexible and belongs to that class of structures which should properly be termed flexible, exactly similar to bridge structures. These flexible motors—that is, motors with drawn steel cylinders and forgings—weave, just as the engine of a steamship weaves. To attempt to hold one against springing is to attempt what is practically impossible. The cylinders of airplane engines should be free to go as they will and not be held on the top in any way. All the block arrangements of cylinders of the sort described are therefore objectionable.

The location of the camshaft down in the crank case with rods coming up to the rocker arms or levers allows each cylinder to be free and does not interfere with its turning in any direction. Also it permits the use of longer levers and a far more accurate adjustment of the



THE GNOME TYPE AIRPLANE ENGINE

timing clearances between the cam and the stem on the long reach rod; but such a reach rod should be a tension rod or else should not be used at all. A push-rod in a place like that seems to be fundamentally wrong. Here is a case of a long column of the thickness of about half one's little finger which is in many cases opening a valve, an exhaust valve, against an internal pressure of 40 lb. per sq.in. and a diameter of two inches or more and in addition overcoming all the inertia of the gear and valve at perhaps 2500 r.p.m. That is not a proper function for a long column, but is a proper function for a tension column or steel wire, and why no one has put a steel wire between the valve stem and camshaft, which will allow the whole structure to go

the way it wants to, instead of abandoning the crank case camshaft, is heard to see. Speaking of valves, the doctor said that it is of no consequence to lift a poppet valve more than one-quarter of its diameter. It is also true that the valve will work better and the volumetric efficiency and mean effective pressure be better, the larger the diameter of the valve and the smaller the lift.

In no case should an inlet valve close sooner than 20 deg. late, and the amount more than 20 deg. late must still be determined experimentally for each machine, because the porting and manifold are different on each machine and no general formula has yet been found. Likewise the exhaust opening must be 45 deg. plus something ahead, the exhaust closing 5 deg. late plus something, etc. The inlet opening is the only period that does not seem to matter.

The doctor is of the opinion that most valve stems are not large enough in diameter. He claims that the stem thickness must bear a logical relation to the disk face, and that the thickness of the face should regularly increase toward the center. If one follows this out, one can easily develop a rational form of valve based on heat flow but not on stresses. The ratio of the conducting area to the heating surface becomes a prime variable, and it is perfectly evident that that ratio ought to be the same all through one piece of metal and ought to be increased when the heat must cross a bridge, as at a stem guide, where there is a film of oil or dead gas, by an amount representing the ratio of thermal resistances.

Pistons Have Been Failures

The ordinary piston as built for aëronautical motors has been a failure, and even in the best motors today next to the exhaust valve the piston is the source of greatest trouble. More accidents and trouble can be traced directly to pistons than to any other single part of the engine structure, and yet pistons have been entirely neglected from this thermal standpoint.

Consider the piston from the viewpoint of heat dissipation and something surprising follows the logic of the analysis. The piston is receiving heat all over the top at a very high rate. Where is the heat going? It must go out through the barrel walls and through the valve stems, through the stem bearings to the water. That is the only place it can go. It is clear that the heat received within a circle drawn concentrically on the head must pass radially outward through a cylindrical surface equal to the circumference of the circle multiplied by the head thickness at that point. The heat received within a larger circle drawn on the head, passing also radially outward, must have a larger head thickness at its circumference. If the temperature is not to get unduly high, then the head thickness must regularly increase from the center outward, so that the metal-conducting area bears a constant ratio to the area of a heat-receiving circle to control the temperature gradient from center to edge. The principle is the same as is used in designing copper electrical conductors to control the voltage drop. It can be shown by a simple equation that the thickness ought to increase on a straight line.

When the heat gets to the edge, it is clear that it must flow down the piston barrel. Therefore there ought to be as much metal behind the first ring as the thickness of the head at the outside of the barrel. Practically no heat can get through the ring, this being a floating member. Then the barrel thickness can be regularly decreased toward the open end to control the temperature gradient from end to end. It is also clear that the more surface there is around the barrel and the better the fit, the easier it is to establish a low temperature gradient between piston barrel and cylinder wall, provided there is sufficient conducting metal in the piston walls, head and barrel. The additional weight is not going to injure the motor, but will permanently have the effect of enabling it to run a longer period of time.

Changes in Crank Case

Speaking of crank cases Dr. Lucke expressed the belief that the next step should be to eliminate the cast aluminum webbed box member entirely and build the whole thing of direct truss form, using nothing but steel. Furthermore, he did not hesitate to say that it was his belief, founded upon the study of practice and on some analyses, that no member of the aëronautical engine that is subjected to heavy stress should be of anything but steel except when that member is subjected to heat-carrying conditions and must be designed for heat carrying rather than stress-resistance, in which case it will be found that there is more than enough metal for stress.

He concluded by saying that the aëronautical engine is emerging from the stage of invention to the stage of design as a light, high-tension, steel structure, consisting of seamless tubing and forged or welded steel parts properly formed in drop-forged dies. Add to that steel stress structure certain members such as the piston, exhaust valve and guide, designed primarily for heat-flow conditions and not for stress. Add to that again certain closing members such as the port for the intake and exhaust which can be very properly cast in aluminum, and the oil crank-case closure, which can be made of anything you please.

The following excerpts are from a paper entitled "Some Notes on High-Pressure Aviation Engines," by Leigh M. Griffith, read before the Aëronautic Section of the Society of Automobile Engineers:

Spark Plug Most Troublesome

"The greatest single source of trouble lies in the spark plugs, which fail not so much because of oiling conditions as from insulation and electrode deterioration due to the combination of high and variable temperatures with gas pressures that also are high and variable. The low-pressure, large volume type of design received an immense impetus from the almost exclusive success of the Gnome engine. The experience obtained by engineers who have been engaged in the design and operation of racing automobiles, the engines of which were at an early date limited in the maximum cylinder dimensions, showed that a satisfactory power plant could be built using much higher gas pressures than the majority of aëronautic engine designers had thought possible. The result was to bring to the engineers working on the aviation engine new men whose experience led them to adopt the high-pressure type of design.

"The author would place the dividing line between the high- and low-pressure engines at about 100 lb. per sq.in. brake m.e.p. four-stroke cycle type, and at 75 lb. for the

two-stroke cycle type. Such a division would place the majority of the modern successful engines, especially of the water-cooled type, in the high-pressure class. Comparatively little had been done in the case of the two-stroke cycle type. It is believed that this latter type offers considerable promise, eventually surpassing the maximum power weight ratio that it is possible to obtain from the four-stroke-cycle type, owing to the apparent ease of obtaining brake mean effective pressure considerably more than half as great as in the case of the four-stroke-cycle engine.

"It may be taken as a general statement that for a given design of engine operating at a given piston speed, it matters little whether the power is developed in large cylinders operating under the low mean pressure, or in small-bore cylinders of high mean pressure. As the weight of the cylinder constitutes a considerable part of the total weight of any aëronautic engine (Winkler, the German authority, places the percentage at about 28, as a water-cooled engine average, which is 5 per cent. greater than for the crank case), it is believed that a consideration of the weight reduction possible for the realization of the adoption of the highest steam pressures would be of considerable interest to designers.

Strong, Thin-Wall Cylinders Needed

"The ideal cylinder would have the thinnest possible wall consistent with the maintenance of a satisfactory degree of rigidity; which would be so designed as to permit the greatest freedom of expansion and contraction under the influence of temperature changes; would have uniform wall thickness throughout; and would be provided with a water jacket of such design as to give a uniform cooling action over all parts of the cylinder wall. The basic idea is to be sure that each unit area of heated surface of the combustion chamber and cylinder walls is separated from a like unit area of uniformly cooled surface by a wall of minimum uniform thermal resistance. Of course in the case of the piston and valve heads this cannot be done, since these members have to conduct the heat they receive largely through their surfaces of contact with the metal of the cylinder in order to transfer it to the cooling water. In the piston this is best accomplished by using a metal of high conductivity and a design that provides ample sectional area to conduct properly the heat from the head to the skirt, from which it passes through the oil films to the cylinder wall.

"Aluminum alloy is by far the best material for use for high-pressure piston construction, mainly because of its high heat conductivity—from three to four times as great as that of either cast iron or steel—but also because of its low specific gravity, which brings about a considerable reduction in the reciprocating weight and in the inertia. Attention has been called to the increased intensity of heat absorption in the case of the high-pressure units, but it would appear that this is almost wholly offset in the case of the piston by the reduced diameter and consequently reduced distance for the heat to travel.

"The best solution of the problem of the proper cylinder construction of an engine working under the highest brake mean effective pressure is that in which the cylinder units are built up in pairs or blocks from alloy steel parts flame-welded together in one integral hole. As showing the weight-reduction possibilities of such construction, the author has built 4⅝ x 7-in. cylinders by this method, from nickel and carbon steel in pairs that weighed only 24 lb. per pair or 9.8 cu.in. displacement per lb. These cylinders are on an engine that on test showed a brake mean effective pressure of more than 116 lb. per sq.in., which is believed to constitute a record for aëronautic engines. The hydrostatic test pressure for inspection was 1000 lb. Moreover, the cylinders were provided with four valves and two spark plugs each, and the jacketing was very complete, extending to the inlet and exhaust flanges, as well as completely around the valve seat and plug bosses.

Power Weight Ratio Greatest with Direct Drive

"The author has become convinced that the greatest power weight ratio can be obtained in the direct-drive engine; this is borne out by the fact that the lightest water-cooled engine so far tested to the author's knowledge is the one incorporating the previously mentioned light-weight cylinders. At the moderate speed of 1325 r.p.m. this engine showed a weight of only 2.6 lb. per brake horsepower. This weight included the magnetos complete with all wiring and plugs, with two carburetors and manifolds, all water headers and a particularly powerful pump.

"Spark plugs are perhaps the greatest source of trouble in the high-pressure engine. The grounded electrodes must be of sufficiently high heat conductivity and must have a sufficiently good thermal contact with the shell to insure that the heat they receive will be carried to the shell and thence to the jacket water. The sectional area of these electrodes should increase from the sparking point to their anchorage in the shell, and they should not be flat or deformed so as to offer a relatively large surface for the absorption of heat from the gas. The greatest improvement is to be looked for along the lines of electrodes with higher heat conductivity, greater sectional area and perhaps thinner insulators having more intimate contact with both the electrode and the shell.

"The author believes that only by the use of multiple valves of moderate diameter can we get the highest volumetric efficiencies and compressions at the same time.

Ignition and Flame Propagation

"Ignition for the high-pressure engines demands far greater attention than the low-pressure engines. The increase of compression increases the rapidity of flame propagation somewhat, and the reduced size of the compression chamber further shortens the time of total ignition, yet to obtain the highest rapidity of flame propagation required for the development of maximum effective pressures, it is necessary to ignite simultaneously the mixture from two or more points as widely separated as possible. This should preferably be done by separate ignition apparatus."

In discussing this paper, E. Planche said that no valve motion in a flying machine engine should be given by a rocker-arm system; that is, the action of the camshaft in opening and closing the exhaust and inlet valves should not be transmitted to the valves through any system that uses a rocker-arm. Airplane engines are working under peculiar and very severe conditions. Their speed is generally higher than that of the commercial engine used in driving automobiles, the power developed is much higher.

Editorials

The Structure of Welds as Applied to Pressure Vessels

There are three divisions of opinion in the present attitude toward welding for pressure vessels, such as boilers, water tanks, ammonia coolers and receivers. First is the manufacturer, who, for cost and production reasons, advocates and applies welding wherever he can and who looks with a sour eye upon all who question the reliability of his welds. Secondly, there are the officers charged with public safety, the insurance companies and others who are just as determined not to allow the use of welds in tension. Happily, the third division comprises those engineers and metallurgists who appreciate all that is desirable in welding, but realize that because of the uncertainty of the strength of the weld the hazards, especially in the case of high-pressure boilers, are too great at this stage of the development of the art to permit the unrestricted application of anybody's weld anywhere.

Up to the present there has been too little of the scientific in relation to welding. The shortest path to success and development of reliable and uniform welds is the study of the microstructure of welded parts by metallurgists skilled in the microstructure of metals, particularly iron and steel, and by physicists.

In welding one is dealing with metal crystals, microscopic bodies made up of molecules, which consist of atoms that in turn, according to modern science, are composed of electrons. When a weld or the metal near the weld breaks, it breaks either at the crystal boundaries or through the crystals. In welding it is the manner of application, the time and the intensity of heat and temperature chiefly that determine whether the crystals of the weld and of the metal near the weld are going to hold together under the stresses imposed by the service to which the joint is to be subjected.

How the crystals are affected is admirably brought out by P. A. E. Armstrong in an article in this issue, probably the most scientific and valuable article yet published, on the microstructure of welds. The practical welder and the engineer cannot but be impressed by the author's contention that the joint before welding should be "tacked" for its entire length and the V filled in by the cast metal in layers, each layer running the full length of the joint and being laid down for the entire length of the joint before the second layer is started. Mr. Armstrong claims that when the V is filled by the welder as he proceeds from one end of the joint to the other, as is done ordinarily, the metal of the weld and the metal near the weld are kept so hot for so long a time that crystal growth of a serious nature is the result, with consequent serious internal stresses. The purpose of filling the V with thin layers of the cast metal is to heat-treat the metal of the weld and of the joint as the weld progresses toward completion. This, of course, amounts to saying that if the welding is properly done there is no need of preheating the plate unless the plate or the material to be welded is much thicker than ordinary tank or boiler plate.

The author says that in testing welds the test should be made in such manner as to anticipate failure through fatigue. The cast structure, which is the welded material, is fairly high in tensile strength; its elastic limit is high and very near to the tensile strength; the elongation is very low, so also is the reduction of area. He contends, therefore, that to submit welded pieces to the usual tests of tension, elastic limit, elongation and reduction of area, does not throw so much light upon the strength of the weld as is desirable. Instead, the shock or vibration test is recommended. This is in accordance with modern opinions. For example, Sir Robert Hadfield, of London, in discussing a paper by Howard J. Stagg before the Steel Treatment Research Club, of Detroit, said in part: "There is no doubt that careless and improper heat treatment, either originally by the maker or afterward by the user, is largely accountable for most of the so-called mysterious failures of boiler plate or other steel for structural purposes, and yet the existence of defects of the peculiar brittle structure are not detected by the usual tensile test." Sir Robert recommended the shock test in preference to the static test for plate and other structural steels. Mr. Armstrong recommends testing welded pressure vessels by submitting them to water pressure and subjecting the joints to a number of blows given by a hammer or by a specially designed machine that will give a uniform blow at a uniform speed, traveling along the joint at a predetermined rate. It is not at all likely that builders of such vessels particularly, and engineers generally, would submit welded tanks to such tests without protest, yet there seems to be no question in the minds of some expert metallurgists that such tests are not only much more valuable in that they tell more than the time-dignified static tests, but that the vibratory tests are sure to come into general use.

Testing the Testers

Maintaining that the individual power plants at county institutions are being operated at a cost in excess of available central-station service, the Public Service Electric Company of New Jersey has secured permission from the Essex County Board of Freeholders to make a series of tests at these different stations to show proof that this situation exists. These tests will be made at the county hospitals at Overbrook and Soho, as well as at the Caldwell penitentiary.

One would be led to question why the county officials are so in the dark as to existing generating costs at the various plants as not to be able to make a comparison with the fixed central-station rates for the service and consumption demanded. Moreover, why is such an important matter, governing possibly ultimate discontinuance

of these plants, left to the jurisdiction of the electric company, an outside interest, unless it be that the county engineering talent is too busily engaged otherwise to investigate the actual workings of power stations that belong to the people. Or is it because the electric company has the better testing equipment, and probably can better the price regardless of operating conditions?

The existing lighting rate of ten cents per kilowatt-hour brings another interesting phase to the matter. When residents of this and other sections of the state are subjected to such a charge for electric service, and which the company continually asserts, year after year, that it cannot reduce consistent with generating costs and net a fair return on investment, why is this extra burden and increased load at peak hours sought at county property at a rate which to all intents will be considerably lower? This would seem worthy of serious reflection—by the people.

Testing the testers might here prove profitable, for with such anxiety on the part of the electric company as to secure special permission to investigate, on its own behalf, this property of the people to ascertain generating costs, why not special permission on the part of the people to investigate a little on their own behalf, to learn exactly what it is costing to generate and distribute electric energy for local service? The State Board of Public Utility Commissioners is apparently finding it difficult to acquire this with accuracy in its rate hearings; possibly the Board of Freeholders, as a matter of reciprocity, can arrange to have tests made at the central station in the interests of the people, for whose cause they engage. While the situation is here localized, it is constantly duplicated in other sections.

The Airplane Engine

Elsewhere in this issue appears constructive discussion of the weaknesses and needs of airplane engines by men prominent in the industry, who have recently read papers on the subject before engineering societies.

Like so many other channels of human effort, the field of aëronautics has been taught much by the war. At, and for some little time after, the beginning of the war, the well-known Gnome rotary engine appears to have been used on most of the airplanes in the service of the Allies. However, with the advent of the heavy battleplanes, reconnaissance machines and bombing planes, requiring long sustained flights and high power, the necessity arose for a reliable water-cooled engine with stationary cylinders. Such a type is now employed on these machines, the rotary engine being used chiefly for fast scouts.

In order to obtain the terrific speed so desirable in the air service on the front, builders have lightened the structure of the machine so that when turning over, and making loops, as pilots must when engaging the enemy, the factor of safety between the pilot and death is reduced to one and a half. It is the engine that contributes seriously to the weight of the 'plane, and in this age of a particular metal for a particular purpose, it seems a reflection upon the designers that they are not able to reduce the engine weight per horsepower more than they have done.

Dr. Charles E. Lucke, of Columbia University, strikes a responsive chord when he advocates for the aëronautical engine a light, high-tensioned steel structure consisting of seamless tubing and forged or welded steel parts, possibly formed in drop-forged dies, for the cylinders and heads; a steel truss structure for the engine face, inclosed in some light sheet material; and piston, exhaust valve and guide designed primarily for heat flow and not for stresses. In this direction the designer must go, it seems, to get lightness with strength. With weight so seriously penalizing the whole 'plane, as it now does, it surely seems ridiculous to have a water jacket of thick cast iron. Of course, there will be difficulties of manufacture in building such engines as Mr. Lucke suggests, and they will likely be more expensive, for a time at least. But they will be worth the effort and the price.

The conditions of flame propagation, as they now exist, seriously penalize the piston speed. The only logical remedy, it seems, is to simultaneously ignite the charge at more than one point in the cylinder.

There is no doubt that one of the greatest sources of trouble is the spark plug. Like the exhaust valve, up to date nothing has been brought forth, according to the discussion, that will withstand the high temperatures met with, which are produced when a machine must remain for hours in the air. While it is encouraging to note that it is hoped that a small amount of air or mixture, admitted direct to the plug cavity in such manner that it washes the center electrode and the insulator, will carry away the heat, it would seem on further consideration that one of the best means of enabling the plug to get rid of its heat would be to increase the cross-sectional area of the heat-conducting element of the plug.

With America practically pledged to furnish thousands of 'planes and aviators—to blind the eyes of the enemy and overcome the present balance in mastery of the air—engine designers here must direct their skill and energies to perfecting the airplane engine. While it is common sense to adopt the best that has been developed abroad, this is not enough. The airplane's weakness is still in the engine and the shortcomings must be eradicated. It will be recalled that in the early days of the automobile European practice was far ahead of our own, and many of our cars and engines were copies of European designs. When the necessity arose, however, it was met by our designers with the result that American practice is the acknowledged leader. Americans were the first to develop the airplane to the successful commercial stage. Let us hope that American engineers will be first to perfect a powerful, reliable and light-weight engine for such important and trying service.

Where is the economy? In 1916 there were sold in this country 27,200,000 twenty-five-watt tungsten lamps, and in the same period 24,000,000 carbon lamps. No doubt the major portion of the carbon lamps were of the fifty-watt sixteen-candlepower variety. This means that fifty per cent. of the energy used by the lamps could have been saved if twenty-five-watt tungsten lamps had been used, and more light would have been obtained. Calculating on the basis of sixteen-candlepower carbon lamps of an average life of five hundred hours and three pounds of coal per kilowatt-hour, about 900,000 tons of coal would have to be burned to supply the carbon lamps with energy, one-half of which could be saved if tungsten lamps were used. All this energy is not supplied from the coal pile, it is true, but even if part comes from our water falls, the public pays the bills just the same.

Correspondence

Turbine for Locomotive Drive

I read with interest the article by Victor W. Zilen, on the turbine for locomotive drive, in the May 1 issue of *Power*. Mr. Zilen proposes a reversible single-stage impulse turbine for each locomotive driver. He also proposes that the turbines be geared directly to the drivers.

I do not think that this would be practical on American railroads. On account of irregularities in the track, American locomotives are built so that each truck-wheel and driver has considerable vertical play, which is resisted by springs. This is obtained, as is shown in Fig. 1, by allowing the journal boxes to play up and down between guides, or pedestal-jaws as they are called. The spring saddle, carrying part of the weight of the locomotive, rests on the journal box.

Unless this arrangement could be abolished, which is not likely, it would be impossible to gear turbines to the drivers, unless the turbines were arranged to move up and down with the drivers, which, of course, would be impractical on account of the complicated construction and the flexible steam and exhaust couplings that this would make necessary. From this it is seen that there must be some form of flexible connection between the drivers and the turbines.

Also, Mr. Zilen's proposal of having a turbine for each locomotive driver would not be practical because of the excessive number of turbines and gears that would be necessary in large locomotives, especially as the modern tendency is to increase the number of drivers, thereby making the locomotive easier on the track.

Moreover, these turbines and gears would have to be placed between the frames, under the boiler, where they could not be easily got at, unless the locomotive was over an inspection pit. The size of the turbines would be limited by a small space available under these circumstances, and of course a small turbine is much less economical at low speeds than a large one.

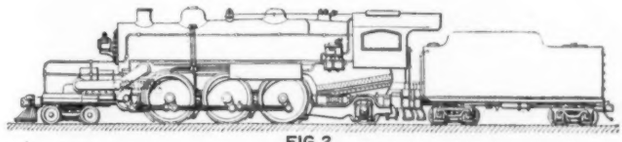
I would suggest having a single large turbine unit placed forward of the boiler and above the frames if necessary, with the turbine shaft parallel to the center line of the engine. This turbine would drive a worm gear, which in turn would drive a gearwheel on a shaft running across the engine, parallel to the driving-wheel axles. This shaft would have cranks on each end, which would drive the driving wheels through side rods. The thrust of the worm gear would be taken up by a thrust bearing on the turbine shaft.

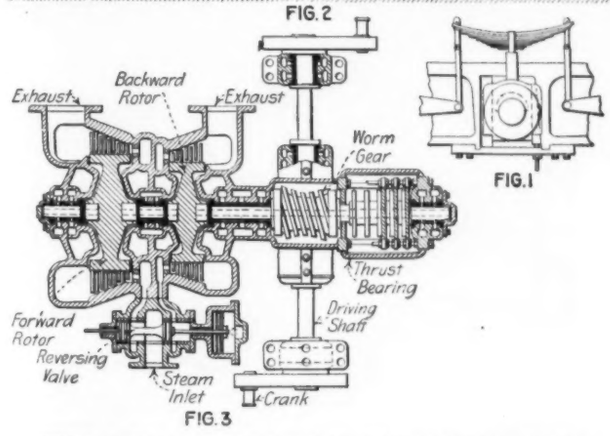
A passenger locomotive with this arrangement is shown in Fig. 2. The turbine, the exhaust pipe, the worm gear, the case over the gearwheel, the driving shaft and crank and the thrust gearing are shown. It is seen that this arrangement would lengthen the total wheelbase of the locomotive, but it would not affect the rigid wheelbase.

Fig. 3 shows a plan of the whole power plant as it would appear from above. The turbine is of the impulse reaction type, having two separate rotors, one for running forward and one for running backward.

It will be noticed that in Fig. 3 the rotor for running backward is smaller than the one for running forward. This is because a passenger locomotive is very rarely required to run backward under full load, and it is not necessary to have the backward rotor as large as the forward rotor. On a freight or drill engine both rotors would be the same size.

To make possible the quick handling required in locomotive service, the admission of steam to the forward and backward portions of the turbine is controlled by a reversing valve, shown in Fig. 3. The reversing valve is operated by a steam cylinder and piston. The admission and exhaust of steam to and from each end of this cylinder is controlled by a valve in the cab of the locomotive. The ends of the reversing-valve chamber are open to the atmosphere to allow any steam that may leak by the ends





FIGS. 1 TO 3. TURBINE FOR LOCOMOTIVE DRIVE

Fig. 1—Spring for truck of locomotive. Fig. 2—Elevation of locomotive driven by one turbine geared. Fig. 3—Section of the proposed turbine

of the reversing valve to escape and also to allow the escape of any steam that may remain in the steam chamber or passages of that portion of the turbine which is cut out. The amount of steam admitted to the turbine is regulated by the regular locomotive throttle valve.

The gears and thrust bearings are inclosed in a case that is kept filled with oil, thus insuring proper lubrication of these parts. By using a worm gear of the right pitch, the turbine, running at its most economical speed, can be made to drive the locomotive at its average speed.

The thrust bearing, as shown in Fig. 3, consists of a series of collars on the turbine shaft bearing against other collars on the bearing frame. The collars on the bearing frame are arranged so that they can be adjusted to take up wear that occurs between them and the shaft collars.

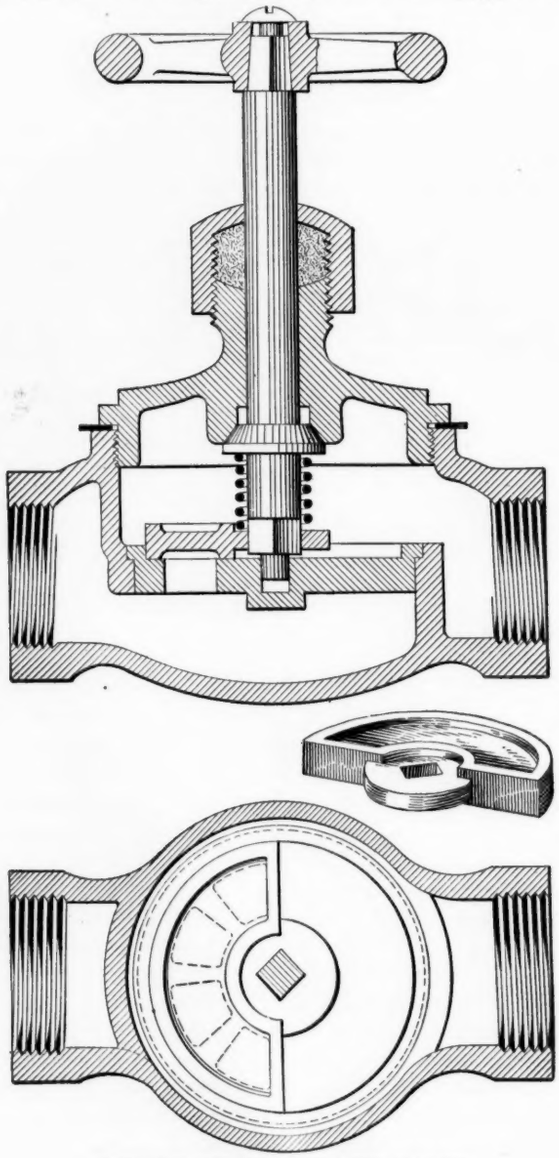
Some difficulty may be experienced when the locomotive is drifting, on account of the gearwheel being unable to

turn the turbine through the worm gear, especially on freight locomotives where the pitch of the worm would be small, but this difficulty could be overcome by admitting enough steam to the turbine to keep it turning while the locomotive is drifting. NICHOLAS M. TRAPNELL.

Elizabeth, N. J.

Multiported Pipe-Line Valve

The accompanying illustration shows a valve that I designed about a year ago after reading in *Power* a letter describing an arrangement of different sizes of valves and piping for controlling the steam supply to a boiler-feed pump to prevent wiredrawing and consequent valve cutting. Upon applying to the Patent Department for patent right on this valve, it was found that a patent had been issued in August, 1900, on a valve designed with



DETAILS OF MULTIPORTED VALVE

the same intentions and practically covering every patentable feature of my valve. I have, however, never seen one of them in use or on the market. Although I could not get a patent and am unable to manufacture my valve, I believe there is a demand for such a valve in almost every plant in the country, and W. Chuisano's letter in *Power*, Apr. 24, verifies this assumption. As the patent I referred to will probably expire next August even if the valve is being produced, which I doubt, would it not be a good idea for any of the valve companies to try it out?

The openings can be used one or all with a minimum cutting effect on the seat, and the fluid cannot cut the face of the disk. The packing collar is made integral with the stem, and if it did not eliminate the use of a fibrous packing, it would lessen the amount needed and allow the packing to be replaced while the valve was in use. Of course I intended to have an indicator of some kind to show the position of the disk. The valve could be opened in either direction, to equalize the wear. The seat cannot be jammed and therefore does away with severe strains to which other valves are subjected.

The square part of the stem that engages the disk is made longer than the thickness of the disk, allowing that member to lift from its seat in case of excessive pressure if the valve was used in a boiler-feed line, as I have seen operators close all feed valves with the throttle valve to pump left open. If this valve was used for this purpose, the pump pressure would, of course, come under the seat.

Trenton, N. J. T. O. VICKERS.

Uniflow Engine Clearance

H. F. Gauss, in his article, "Steam Engineer's License Examinations," in the issue of May 15, page 661, is hardly justified in his statement that the uniflow engin operating noncondensing fails to show good economy when the clearance volume is large. My opinion is that the amount of the clearance volume is of less importance than he would have us believe, but rather that it is the amount of clearance surface that determines the cylinder condensation.

It is conceded that a uniflow engine with small clearance cannot operate noncondensing without relief or auxiliary exhaust valves, but by using a piston with concave heads the clearance volume can be increased to any extent desired, while the clearance surface is increased but slightly. For example, a piston with the heads so concaved as to form half of a hollow sphere would not cause the clearance surface to be increased more than 20 per cent. over that of an engine having only 2 per cent. clearance volume with a straight-faced piston.

The objection may properly be raised that this type of piston with the accompanying large clearance volume prevents the engine from being operated condensing, but it is comparatively seldom that an engine is operated under both conditions. If operating condensing, the use of relief valves to protect the engine from excessive compression, in case the vacuum is lost, is free from objection, but while overcoming the compression problem, it brings in the question of the reliability and certainty of operation in case the vacuum fails. I know of but one instance, however, where the fracture of a cylinder head was attributed to the failure of the auxiliary exhaust valves to work, but it does constitute an objection.

I do not believe any engineer need worry about the difficulties or condemn the valveless uniflow engine until he has compared the actual operating economies of both types. Many are of the opinion that the economy of the uniflow engine is due in the main to the steam jacketing of the heads, but that a counter flow engine completely steam-jacketed, including the interior of the

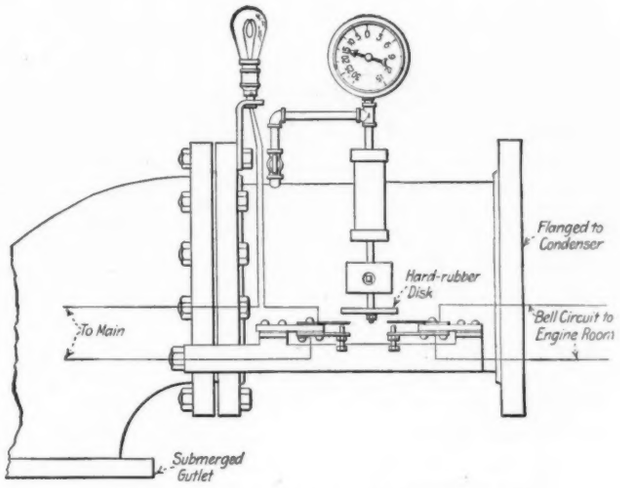
piston, will show better economy than that guaranteed by the uniflow builders. This is proved by a number of tests, and I understand that at least one engine builder is now offering such a steam-jacketed counter-flow engine.

L. H. MORRISON.

Dallas, Tex.

Telltale on Condenser Line

The illustration shows an indicating and alarm-bell ringing device used on the 10-in. circulating-water outlet of a 1500-hp. vertical condenser. The operation of the device is as follows: When the pump is started and full water flow and vacuum is established, the 1½-in. plug piston in the brass cylinder is raised, carrying with it the weighted rod and rubber disk, opening the bell and light circuits. This position is maintained as long as the vacuum is good and the circulating pump is delivering enough water. If for any reason the flow of water is interrupted, the weighted piston falls, allowing the rubber disk to close both circuits.



CIRCUITS CLOSE IF PIPE IS NOT FULL

This combination enables the engineer to locate the trouble and reëstablish normal conditions before the condenser and air pump get overheated. If time had permitted the making of patterns, etc., a neater-looking design could have been produced, but this one serves the purpose first-rate.

P. J. KEATING.

Streator, Ill.

Pump Exhausting Into Suction

In the issue of May 15, page 672, J. C. Hawkins asked the opinion of other engineers regarding connecting a pump exhaust to the suction line.

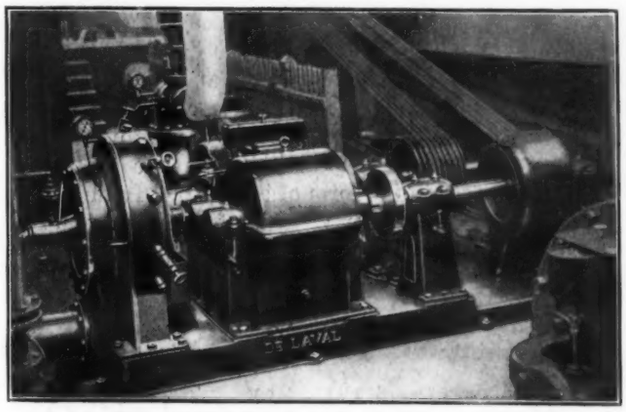
I have connected the exhausts of many mine pumps in this way with success. I place a tee close to the pump and pipe the exhaust through the tee (a little more than past the side opening). The part of the suction next to the pump is enlarged to where the steam exhausts into it. The pump must be started first with the exhaust free until the flow of water has been established, when the exhaust can be turned into the suction.

Montesano, Wash. E. M. KEYS, JR.

Steam Turbines for Mill Drive

The article on "The Evolution of the Steam Turbine," by John A. Stevens, abstracted in the issue of May 22, page 712, gives the impression that the first geared steam turbine used for mechanical drive in mills was a 700-hp. unit installed in 1913 in a jute mill in Calcutta, India.

In a paper read by E. B. Storie before the Manchester Association of Engineers and reprinted in *The Engineer* of Jan. 28, 1910, it was stated that although the application of the steam turbine for driving textile mills dated no further back than nine or ten years, there were at that time not fewer than 480 turbines in use for that purpose, aggregating 300,000 hp., of which about 3 per cent. were for mechanical driving. Storie gave a table listing 118 Swedish De Laval, 33 English De Laval and 80 French De Laval turbines, a number of which we know to have been geared turbines adapted for belt or rope drive.



SINGLE-STAGE TURBINE DOUBLE-GEARED TO ROPE TRANSMISSION

The accompanying photograph shows a De Laval single-stage double-geared steam turbine in use driving a cotton mill with rope transmission. This unit developed 375-b.hp. at 750 r.p.m. As it was shown in a De Laval catalog copyrighted in 1911, it antedates, by a number of years at least, the Calcutta unit mentioned by Mr. Stevens, also the 525-hp. unit which Mr. Stevens mentions having been started in 1914 in a paper mill at Bolton, England.

There might also be mentioned the multistage mixed-flow turbine and double-gear reduction that was supplied in 1914 to the Carpenter Steel Co., Reading, Penn. (See *Power*, July 20, 1915, p. 74). This turbine is, we believe, both earlier and larger than some of those mentioned by Mr. Stevens as having been pioneers in the line of mechanical drives for mills and was the first in the United States for rolling-mill service.

New York City. GEO. H. GIBSON.

Freight Rates on Fuel—The Vandalia Railroad Co. and other roads established a freight rate of 75c. per ton on coal from mines to Kokomo, in the case of coal used for general steam purposes, and a rate of 65c. for manufacturers whose outbound products were carried by the coal-carrying road. In these circumstances it is held by the Indiana Supreme Court that an electric-power company engaged in supplying power to factories at Kokomo is entitled to the benefit of the lower rate on that portion of its coal used in producing power furnished to manufacturers who ship their products out over the same road. (Vandalia Railroad Co. vs. Public Service Commission, 114 Northeastern Reporter, 412).

Relative Benefit of High Vacuum

The second editorial on page 389 in the issue of Mar. 20 starts off with, "Since the advent of the turbine has made important the maintaining of a high vacuum,—" then the next sentence begins with "Engineers accustomed to reciprocating-engine conditions, where a vacuum of 26 in. was all that was required,—."

These two statements of a great truth will set many an engineer thinking why such should be the case, and he will wish that the purpose of the editorial had involved an answer to his "why." Personally it makes me think of the hard time I had locating the answer, and I would like to tell what I found and how I found it.

Three years ago when up for an examination I was asked: "Why does it pay to spend large sums of money for special condensing machinery in order to get an extra high vacuum for a turbine—and not pay to do the same for a reciprocating engine?"

I was "stuck solid" and admitted it. I went home and got busy with my books for the next few evenings, but no answer was to be found. I went to an engineer friend and he did not know; then to others (five in number), all of whom I considered my superiors, and they too were "stalled"; none of them could tell me. These sources of information failing, I next thought of the turbine manufacturers and wrote for catalogs. In one of them I found my answer in the form of an abstract from a paper on "Steam Engine Economy" by Prof. J. E. Denton, read before the mechanical section of the International Congress of Arts and Science at St. Louis on Sept. 23, 1904, as follows:

One of the most important advantages of the steam turbine as compared with the reciprocating engine is that it can be made to efficiently utilize all the available energy incident to the use of vacuum. The most economical piston engines, which, as we have seen, expand steam about 39 times, release their steam at the end of the stroke at 6 lb. absolute pressure, but they exhaust into a condenser, which with a vacuum of 28 in. contains a pressure of about 1 lb. absolute per square inch. If the expansion could continue until the pressure of 1 lb. was attained before exhaust occurred, considerably more work would be obtained from the steam. This, however, cannot be done in piston engines for two reasons: First, because the low-pressure cylinder would have to be about five times greater in volume, which is commercially impracticable; and second, because the velocity of exit through the largest exhaust ports possible is so great that the frictional resistance of the steam makes a pressure from 1 to 3 lb. higher than the condenser pressure in the best engines of ordinary piston speed.

The steam turbine expands its steam to the pressure of the exhaust chamber, and as unlimited escape ports can be provided from this chamber to a condenser, it follows that the turbine can practically expand its steam to the pressure of the condenser.

I am not finding fault with the editorial or editor either, but I fear that what is just an everyday remark to turbine engineers is not common knowledge to us reciprocating-engine operators. R. MANLY ORR.

Vancouver, B. C., Canada.

Steam for Gas-Producer

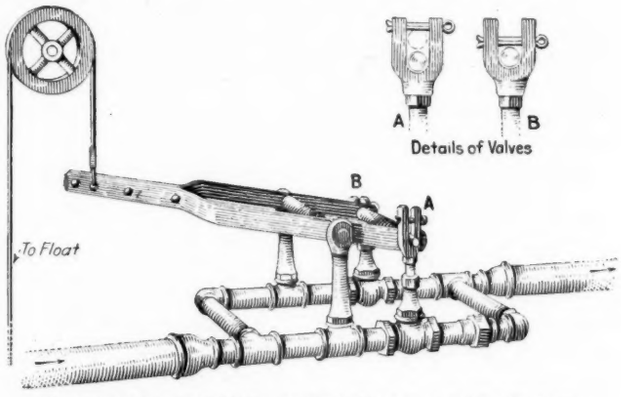
I am pleased to see an occasional item of interest to producer gas-plant operators, as in the issue of May 29, page 749. I am operating two up-draft suction producers fired with anthracite pea coal, and each producer is equipped with a thermometer at the top of the saturator. I have had the best success carrying a temperature from 129 to 136 deg. F. (54 to 58 deg. Centigrade).

By regulating the mixing valve to keep the governor from hunting, I am able to eliminate preignition, which had previously caused a couple of shutdowns a day.

Cooperstown, N. Y. R. L. HAYDEN.

Float-Operated Regulating Valves

For years I had trouble with the piston valve on a float-type pump regulator. The valve would score, and when the returns fell off after 6 p.m. to the minimum, the leakage of the valve was sufficient to empty the receiver and cause the pump to pound. A new valve was put in, and in a short time it was cut as badly as the first, so a valve-operating rig was made of fittings and has been in service for some years with perfect success.



TWO SMALL VALVES IN PLACE OF A LARGE ONE

The illustration shows how it was made. The steam feed is one inch, but half the area of a 1-in. pipe is sufficient at all times to run the pump. Two ½-in. valves were used so that a larger valve would not be open too little and score the seat. Steam is admitted on top of the valves, which keeps them on their seats. Lost motion in one valve yoke allows the other valve to open first, and if only a slight opening is required one valve is sufficient, but should a slug of water come along the float is raised high enough to lift the second valve from its seat. New disks can be put in easily in twenty minutes if necessary, but during the last winter the pump operated 24 hours daily and no new disks were necessary. Of course the threads are turned from valve stems, and it is necessary to have the use of machine tools for the construction of the regulator, but it eclipses any regulating valve we have ever had. C. J. WILDER.

Auburn, Mass.

Handy Ratchet Brace

Referring to James E. Noble's letter in the issue of May 8, page 637, describing a "Handy Ratchet Brace for Pipes," the illustration does not show how the brace is prevented from turning about the point where the horizontal bar is bolted to the clamp.

The triangular part is all right, but the portion of the horizontal bar between the clamp bolt and point of intersection of the vertical and horizontal bar is unsupported. The brace will either turn about the clamp bolt, or if one-half of the clamp and the horizontal bar are made in one piece, the bar will bend.

Kewanne, Ill. FRANZ SZABO.

Inquiries of General Interest

Fusible Plugs in Horizontal Return-Tubular Boilers—How should soft or fusible plugs be placed in horizontal return-tubular boilers? J. O.

In a horizontal return-tubular boiler the fusible plugs should be located in the rear head, not less than 2 in. above the upper row of tubes, the measurement to be taken from the line of the upper surface of tubes to the center of the plug, which should project through the sheet not less than one inch.

Fusible Plug in Cross-Drum Water-Tube Boiler—Where is the fusible plug placed in the B. & W. cross-drum type boiler? J. D.

The fusible plug or plugs are placed in the under side of the steam drum toward the front of the boiler, on a radial line that usually is about 45 deg. around from the lowest side of the drum, though not less than 6 in. above the bottom of the drum, in the path of the products of combustion, and projecting through the sheet not less than 1 in.

More Compression from Lengthening Exhaust-Valve Rod—On a Corliss engine is more compression obtained by lengthening or by shortening the exhaust-valve rods? H. H.

In the ordinary Corliss valve gear, the valves are opened by the wristplate pulling the valve rods and closed by thrusting the valve rods. Hence, lengthening the exhaust-valve rod would cause the port to be closed earlier and there would be a larger volume of exhaust steam compressed into the clearance space, resulting in compression of the exhaust to a higher pressure. It should be borne in mind, however, that lengthening of the exhaust valve rod also has the effect of adding lap to the valve, thereby causing later opening for release after expansion has taken place.

Overstroking of Duplex Boiler-Feed Pump—One side of a duplex steam pump used for boiler feeding runs smoothly, but the other side strikes at both ends of the stroke. How can the trouble be remedied? W. V. H.

There should be less lost motion in the gear of the steam valve of that side of the pump that overstrokes, so the valve will close earlier in the stroke. If the regular lost-motion mechanism provided for the steam valve consists of a single nut on the valve stem, operating between two lugs on the valve, the lost motion can be reduced and the valve can be made to close earlier by providing a thicker nut. If the lost motion is obtained by nuts or other stops on the valve rod outside of a lug or other projection on the valve, the distance between the stops and striking points on the valve should be reduced.

Use of Muddy Feed Water—What effect has muddy feed water on a boiler, and how can injurious effects be prevented? A. P.

When muddy feed water is used, the heavier material in suspension settles in the water-legs of the boiler or wherever the circulation is sluggish, and if the mud is allowed to harden on the heating surfaces, the heat-transmitting power of the boiler material is lowered and it may become blistered or burned. If possible, the feed water should be clarified of mud before use by means of settling tanks. If these cannot be employed, the boiler should be provided with a mud-collecting chamber connected with the lowest part of the boiler, from which the mud can be discharged at intervals by opening a blowoff valve.

Grounding Current Transformers—Why should one side of the secondary of a current transformer be grounded, while an instrument potential transformer is not grounded? R. C.

There is no more reason for not grounding one side or the neutral of the secondary of the one than the other. One of the main reasons for grounding the low-voltage winding on any transformer is that in case of an accidental insulation breakdown between the primary and secondary windings, there will be no danger of a person receiving a high-voltage shock from the low-voltage winding to ground. The metal mountings on the transformers should also be grounded for the same reason.

Dielectric Strength and Insulation Resistance—What is the difference between dielectric strength and insulation resistance? T. A.

All known electrical insulating materials when subject to an electrical potential will allow an electric current to flow through them. This current of course is of very small value. The voltage applied across the insulation divided by the current gives the insulation resistance in ohms, or in other words, insulation resistance is the resistance measured in ohms or megohms offered by any insulated material to the flow of electric current. Dielectric strength is the property of an insulating material to withstand puncture by a high voltage, and is measured in volts per millimeter thickness. However, materials that have high insulation resistance do not necessarily have high dielectric strength. Experience has shown that some insulators that show high insulation resistance on low voltage, when subject to a high-potential test (dielectric test) break down on comparatively low voltages.

Slip of Pump—What is meant by the term "slip" of a pump, how is it measured and what should be the percentage of slip of a pump in good working order? N. C.

On account of defective piston packing, leaky stuffing-boxes or valves, delayed closing of the valves and the amount of air carried into the pump by the body of water handled, the actual volume of water discharged, generally stated in gallons per minute, is less than the plunger displacement, expressed in the same unit. This loss, stated in percentage of the displacement, is called slip. In a new pump handling water that is reasonably free from air, the slip should not exceed 2 per cent. In pumps handling large quantities of water, the percentage of slip is frequently as little as ½ per cent., and in some instances there is no slip or even negative slip from the tendency of the water to continue to flow after the plunger has come to rest. The loss from slippage depends on the condition of the pump, the pressure and the speed, and as it is mainly due to leakage of valves and packing that is practically constant for all speeds and considered as a percentage of piston displacement, the slip decreases for an increase of plunger speed. For testing the slip of a direct-acting steam pump, close the discharge valve and admit sufficient steam pressure to produce the normal discharge pressure. The piston speed multiplied by 100 and divided by the normal piston speed will be the approximate percentage of slip for the normal speed that is assumed. For average working conditions this should not exceed 10 per cent.

Power Developed by Current Waterwheel—What power would be developed by a current waterwheel having paddles or floats 3 x 6 ft., operated in a stream flowing at the rate of 220 ft. per min.? G. G.

The cross-sectional area of water intercepted by the floats would be 3 × 6, or 18 sq.ft., and having a current velocity of 220 ft. per min., or 3.66 ft. per sec., the volume of water intercepted would be 18 × 3.66 = 65.88 cu.ft. per sec. and the weight of water presented per second would be 65.88 × 62.3 = 4104.3 lb.

The energy developed by the water is given by the formula for kinetic energy,

$$E = \frac{W}{2g} v^2$$

in which

E = Foot-pounds per second;
W = Weight in pounds;
2g = 64.4;
v = Velocity, feet per second,

and by substitution, the kinetic energy of the water is found to be

$$E = \frac{4104.3}{64.4} \times (3.66)^2 = 853.72 \text{ ft.-lb. per sec., or } 853.72 \div 550 = 1.55 \text{ gross horsepower.}$$

With the floats plane and radial, and spaced at distances equal to their radial length, Poncelet found that the efficiency of wheels of this kind was greatest when the velocity of the center of a float was equal to one-half the velocity of the current, the efficiency at that speed being 0.40. Hence the greatest power developed by the wheel would be 40 per cent. of 1.55, or about 0.62 hp. The net power realized would depend on the character of gearing employed.

[Correspondents sending us inquiries should sign their communications with full names and post office addresses. This is necessary to guarantee the good faith of the communications and for the inquiries to receive attention.—Editor.]

Revisions to the Boiler Code

SYNOPSIS—Many interesting revisions to the Boiler Code are given. Constructive criticism is called for before the revisions go to the Council of the American Society of Mechanical Engineers for approval.

On Dec. 8 and 9, the last meeting of the Boiler Code Committee of the American Society of Mechanical Engineers at which all interested were invited to attend, changes in the Boiler Code were proposed. Following are some proposed revisions to the Code; and more to be published later, according to the June "Journal" of the society. Those interested are urged now to present to the committee constructive comment on these revisions. The Boiler Code may be had by applying to the secretary to the Code Committee, American Society of Mechanical Engineers, 29 West 39th St., New York City. The cost is 80c. to nonmembers and 40c. to members.

Only such portions of the Code as are subject to revision are here presented, with references to their proper paragraph numbers.

Page 8, Par. 12. Change to read as follows: Cast iron shall not be used for nozzles or flanges attached directly to the boiler at any pressure or temperature. Cast iron shall not be used for boiler and superheater mountings such as connecting pipes, fittings, valves and their bonnets, for steam temperatures of over 450 deg. F.

Par. 13. Change to read as follows: Water-leg and door-frame rings of vertical fire-tube boilers, and of locomotive and other type boilers, shall be of wrought iron or steel, or cast steel of Class A or Class B grade, as designated in the Specifications for Steel Castings. The OG or other flanged construction may be substituted.

Page 9, Par. 19. Change to read as follows: The minimum thickness of butt straps shall be given as in Table 1. Intermediate values shall be determined by interpolation. For plate thicknesses exceeding 1¼ in., the thickness of the butt straps shall be not less than three-quarters of the thickness of the plate.

Page 11, Par. 25. Make the following changes: Strike out the word "Copper" in the table, and under "Firebox" strike out "not over 0.05 per cent."

Page 31, Par. 123. Par. 123a: Change the percentage of yield point from 0.6 tensile strength to 0.5.

Page 34, Par. 141. Change the percentage of yield point from 0.6 tensile strength to read 0.5 tensile strength.

Page 40, Par. 164. Make the following change: After the word "knobbled" insert a comma.

Par. 167. Change to read as follows: Flange Test. a. For tubes not more than 6 in. diameter a test specimen not less than 4 in. in length shall have a flange turned over at right angles to the body of the tube without showing cracks or flaws. This flange as measured from the outside of the tube shall have a width of from ⅛ in. to ½ in. The width between these limits to be not less than 10 per cent. of the outside diameter of the tube. For tubes more than 6 in. diameter the flange test is not required.

b. In making the flange test, the flaring tool and die block as shown in Fig. 7, may be used.

Page 41, Par. 169. Change to read as follows: Hydrostatic Tests. Tubes under 5 in. in diameter shall stand an internal hydrostatic pressure of 1000 lb. per sq.in. and tubes 5 in. in diameter or over, an internal hydrostatic pressure of 800 lb. per sq.in., provided that the fiber stress does not exceed 16,000 lb. per sq.in., in which case the test pressure shall be determined by the following formula,

$$P = \frac{32{,}000}{D} \times t$$

where t is the wall thickness in inches, D is the inside diameter in inches. Lapwelded tubes shall be struck near both ends, while under the test pressure, with a 2-lb. steel hand hammer, the blow to be equivalent to 2 lb. falling 2 ft.

Page 42, Par. 174. Change to read as follows: Workmanship. Finished tubes 3½ in. or under in outside diameter shall be circular within 0.02 in. and the mean outside diameter shall not vary more than 0.015 in. from the size ordered. For tubes over 3½ in. in outside diameter, these variations shall not exceed 0.5 per cent. of the outside diameter. All tubes shall be carefully gaged with a B.w.g. gage and shall not be less than the gage specified. Tubes on which the standard slot gage, specified, will go on tightly at the thinnest point, will be accepted. The length shall not be less, but may be 0.125 in. more than that ordered.

Page 44, Par. 184. Change to read as follows: Circumferential Joints. a. The strength of circumferential joints of boilers, the heads of which are not stayed by tubes or through braces, shall be at least 50 per cent. of that required for the longitudinal joints of the same structure.

b. When 50 per cent. or more of the load which would act on an unstayed solid head of the same diameter as the shell is relieved by the effect of tubes or through stays in consequence of the reduction of the area acted on by the pressure and the holding power of the tubes and stays, the strength of the circumferential joints in the shell shall be at least 35 per cent. of that required for the longitudinal joints.

c. In circumferential joints of horizontal return-tubular boilers the shearing strength of the rivets shall be not less than 50 per cent. of the full strength of the plate corresponding to the thickness at the joint.

Page 45, Par. 186. Change to read as follows: Welded joints. The ultimate strength of a joint which has been properly welded by the forging process shall be taken as 28,500 lb. per sq.in., with steel plates having a range in tensile strength of 47,000 to 55,000 lb. per sq.in. Autogenous welding may be used in boilers in cases where the strain is carried by other construction which conforms to the requirements of the Code and where the safety of the structure is not dependent upon the strength of the weld. Autogenous welding shall not be used in place of calking or girth joints.

NOT OVER 12-FT. COURSE FOR LAP JOINTS

Par. 187. Change to read as follows: Riveted Longitudinal Joints. The riveted longitudinal joints of a shell or drum which exceeds 36 in. in diameter, shall be of butt and double-strap construction. This rule does not apply to the portion of a boiler shell which is staybolted to the firebox sheet.

Par 190. Change to read as follows: In horizontal return-tubular boilers with lap joints no course shall be over 12 ft. long. With butt and double-strap construction, longitudinal joints of any length may be used provided the tension-test specimens are so cut from the plate that their lengthwise direction is parallel with circumferential seams of the boiler and the tests meet the standards prescribed in the specifications for boiler plate steel.

Par. 194. Change the first section to read as follows: Domes. The longitudinal joint of a dome 24 in. or over in diameter shall be of butt and double-strap construction irrespective of pressure. When the maximum allowable working pressure exceeds 100 lb. per sq. in., the flange of a dome 24 in. or over in diameter shall be double-riveted to the boiler shell.

Page 49, Par. 195. Add after notation following formula: Where two radii are used, the longer shall be taken as the value of L in the formula. Add at the end of Par. 195: When a dished head has a manhole opening, the thickness as found by these Rules shall be increased by not less than ⅛ in. over that called for by the formula.

Page 50, Par. 199. Add the following to the list of values of C: C = 150 for stays screwed through plates or made a taper fit and having the heads formed on the stays before installing them and not riveted over, said heads being made to have a true bearing on the plate and the diameter of the heads being not less than 1.4 times the diameter of the stays.

DOUBLING PLATES FOR STRENGTHENING BOILERS

Revise matter following list of values of C to make it read as follows: If flat boiler plates not less than ⅜ in. thick are strengthened with doubling plates securely riveted thereto and having a thickness of not less than ⅜ t, then the value of t in the formula shall be three-quarters of the combined thickness of the boiler plate and doubling plates, but not more than 1⅛ times the thickness of the boiler plate, and the values of C given above may also be increased 15 per cent. When two sheets are connected by stays and but one of these sheets requires staying, the value of C is governed by the thickness of the sheet requiring staying.

Par. 200. Change to read as follows: Stay-bolts. The ends of screwed stay-bolts shall be riveted over or upset by equivalent process. Stay-bolts must be hollow or the outside ends of solid stay-bolts shall be drilled with a hole at least $\frac{5}{16}$ in. diameter to a depth extending at least ½ in. beyond the pressure side of the plates, except on boilers having a grate area not exceeding 15 sq.ft., or the equivalent in gas- or oil-fired boilers, where the drilling of the stay-bolts is optional. Flexible stay-bolts of either the jointed or ball-and-socket type need not be drilled.

Par. 201. Add the following: If the outstanding legs of the two members are fastened together so that they may act as one member in resisting the bending action produced by the load on the rivets attaching the members to the head of the boiler, and provided that the spacing of these rivets attaching the members to the head is approximately uniform, the members may be figured as a single beam uniformly loaded and supported at the points where the through braces are attached.

Page 51, Par. 203. Change to read as follows: a. The maximum spacing between centers of rivets or between the edges of tube holes and the centers of rivets attaching the crowfeet of braces to the braced surface, shall be determined by the formula in Par. 199, using 135 for the value of C.

b. The maximum distance between the edges of tube holes and the centers of other types of stays shall be determined by the formula in Par. 199 using the value of C given in Par. 199, which applies to the thickness of plate and type of stay used.

c. The maximum spacing between the inner surface of the shell and lines parallel to the surface of the shell passing through the centers of the rivets attaching the crowfeet of braces to the head shall be determined by the formula in Par. 199, using 175 for the value of C.

d. The maximum distance between the inner surface of the shell and the centers of braces of other types shall be determined by the formula in Par. 199, using a value of C equal to 1.3 times that value of C in Par. 199, which applies to the thickness of plate and type of stay as therein specified.

e. In applying these rules and those in Par. 199 to a head or plate having a manhole or reinforced opening, the spacing applies only to the plate around the opening and not across the opening.

Under Table 3 insert: For the application of Pars. 205, 206 and 207, see Appendix (Par. . . .) (illustrations in Case No. 13).

Page 53, Par. 214. Change to read as follows: Areas of Segments of Heads to be Stayed. The area of a segment of a head to be stayed shall be the area enclosed by lines drawn 2 in. from the tubes and a distance d from the shell as shown in Figs. 13 and 14. The value of d shall be the larger of the following values but not less than 3 in.

(Note:—Dimensions marked 3" in Figs. 13 and 14 to be changed to d.)

(1) d = the outer radius of the flange not exceeding eight times the thickness of the head

(2) $$d = \frac{5 \times t}{\sqrt{P}}$$

Where

d = distance in inches
t = thickness of head in sixteenths of an inch
P = maximum allowable working pressure in lb. per sq.in.

Par. 215. Change to read as follows: When drum heads of water-tube boilers are 30 in. or less in diameter and the tube plate is stiffened by flanged ribs or gussets, no stays need be used if a hydrostatic test to destruction of a boiler or unit section built in accordance with the construction, shows that the factor of safety is at least 5.

Par. 216. Change to read as follows: Stays shall be used in the tube sheets of a fire-tube boiler if the distance between the edges of the tube holes exceeds the maximum pitch of stay-bolts for the corresponding plate thickness and pressure given in Table 3. That part of the tube sheet which comes between the tubes and the shell need not be stayed if the nearest tangent common to two tube holes when measured on any radius of the tube sheet that intersects the tangent between the holes does not exceed this maximum pitch by more than 3 in. The tube holes to which a common tangent may be drawn in applying this rule shall not be at a greater distance from edge to edge than the maximum pitch referred to.

Page 54, Par. 218. Add the following: The distance in the clear between the bodies of the braces, or of the inside braces where more than two are used, shall not be less than 10 in. at any point.

Par. 220. Change to read as follows: a. The full pitch dimensions of the stays shall be employed in determining the area to be supported by a stay and the area occupied by the stay shall be deducted therefrom to obtain the net area. The product of the net area in square inches by the maximum allowable working pressure in lb. per sq.in. gives the load to be supported by the stay.

b. Where stays come near bounding surfaces and special allowances are made for the spacing, the load to be carried by such stays shall be determined by neglecting the added area provided for by these special allowances. For example, if the minimum pitch by Table 3 would make a stay-bolt come 6 in. from the edge of the plate and a special allowance would make it come 7 in., the distance of 6 in. should be used in computing the load to be carried.

c. The maximum allowable stress per square inch net cross-sectional area of stays and stay-bolts shall be as given in Table 4.

d. The length of the stay between supports shall be measured from the inner faces of the stayed plates. The stresses are based on tension only. For computing stresses in diagonal stays, see Pars. 221 and 222.

Table 4. Add line after item b in table to read as follows:

c. Steel through stays exceeding 1¼ in. diameter	10,400	9000

Change present item c in table to item d.

Page 55, Par. 223. Change to read as follows: Diameter of Pins and Area of Rivets in Brace. All rivet and pin holes shall conform with the requirements in Par. 253 and the pins shall be made a neat fit. To determine the sizes that shall be used proceed as follows:

1. Determine the "required cross-sectional area of the brace" by first computing the total load to be carried by the brace, and dividing the total load by the values of stresses for unwelded stays given in Table 4.

2. Design the body of the brace so that the cross-sectional area shall be at least equal to the "required cross-sectional area of the brace" for unwelded braces. Where the braces are welded, the cross-sectional area at the weld shall be at least as great as that computed for a stress of 6000 lb. per sq.in. (see Table 4).

3. Make the area of pins to resist double shear at least three-quarters of the "required cross-sectional area of the brace."

4. Make the combined cross-section of the eye at the side of the pin (in crowfoot braces) of at least 25 per cent. greater than the "required cross-sectional area of the brace."

5. Make the combined cross-sectional area of the rivets at each end of the brace at least one and one-quarter times the "required cross-sectional area of the brace."

6. Design each branch of a crowfoot to carry two-thirds the total load on the brace.

SECTIONAL AREAS FOR BRACES

7. Make the net sectional areas through the sides of the crowfoot, tee irons, or similar fastenings at the rivet holes at least equal to the required rivet section. That is, at least equal to 1¼ times the "required cross-sectional area of the brace."

8. Make the cross-sectional areas through the blades of diagonal braces where attached to the shell of the boiler at least equal to the required rivet section, that is, at least equal to 1¼ times the "required cross-sectional area of the brace."

Page 58, Par. 231. Change to read as follows: Maximum Allowable Working Pressure on Truncated Cones. a. Upper combustion chambers of vertical submerged tubular boilers made in the shape of a frustum of a cone when not over 38 in. diameter at the large end, may be used without stays if figured by the rule for plain cylindrical furnaces (Par. 239) making D in the formula equal to the diameter at the large end.

b. When over 38 in in diameter at the large end, that portion which is over 30 in. in diameter shall be fully supported by stay-bolts or gussets to conform to the provisions for staying flat surfaces. In this case the top row of stay-bolts shall be at a point where the cone top is 30 in. or less in diameter.

In calculating the pressure permissible on the unstayed portion of the cone, the vertical distance between the horizontal planes passing through the centers of the rivets at the cone top, and through the center of the top row of stay-bolts shall be used as L in Par. 239, and D in that paragraph shall be the inside diameter at the center of the top row of stay-bolts.

Page 61, Par. 239. Change first paragraph of Par. 239 to read as follows: Plain Circular Furnaces. Unstayed furnaces more than 18 in. diameter when riveted or of seamless construction or such furnaces when lapwelded by the forging process shall have walls not less than $\frac{7}{16}$ in. thick. The maximum allowable working pressure for such furnaces shall be determined by one or the other of the following formulas:

Make definition of L read as follows:

L = total length of furnace between centers of head rivet seams (not length of a section), in.

Insert after the formula and just preceding the example in Par. 239, eliminating the present sentence:

Where the furnace has a riveted longitudinal joint, it may be of the lap type for inside diameters not exceeding 30 in. but shall be of butt and strap construction for inside diameters exceeding 30 in. The efficiency of the joint shall be greater than

$$\frac{P \times D}{1250 \times T}$$

Page 64, Par. 246. Change Par. 246 to Par. 246a.

Par. 247. Change Par. 247 to Par. 246b.

Change Par. 247 to read as follows: Where it is impossible to calculate with a reasonable degree of safety the strength of a boiler structure or any part thereof, a full-sized sample shall be built by the manufacturer and tested to destruction in the presence of the Boiler Code Committee or a representative of the Boiler Code Committee appointed to witness such test.

Par. 250. Change to read as follows: A fire-tube boiler shall have the ends of the tubes substantially rolled and beaded, or rolled and welded, at the firebox or combustion chamber end.

Page 65a, Par. 253. Change to read as follows: Riveting. Rivet holes shall be drilled full size, or they may be punched not to exceed ¼ in. less than full diameter for material over $\frac{5}{16}$ in. in thickness and ⅛ in. less than full diameter for material not exceeding $\frac{5}{16}$ in. in thickness, and then drilled or reamed to full diameter.

Par. 255. Add the following: Forms of rivet heads that will be acceptable are shown in Fig. 17a.

Page 65b, Par. 257. Change to read as follows: Calking. The calking edges of plates, butt straps and heads shall be beveled. Every portion of the sheared surfaces of the calking edges of plates, butt straps and heads shall be planed, milled or chipped to a depth of not less than ⅛ in. Calking shall be done with a round-nosed tool.

Page 67, Par. 266. Change to read as follows: A vertical fire-tube boiler, except boilers of steam fire engines, or boilers 24 in. or less in diameter, shall have not less than seven handholes, located as follows: Three in the shell at or about the line of the crown-sheet; one in the shell at or about the water line or opposite the fusible plug when used; three in the shell at the lower part of the water-leg. A vertical fire-tube boiler, submerged-tube type, shall have two or more handholes in the shell, in line with the upper tube sheet. All boilers 24 in. or less in diameter shall have at least one opening for inspection and one opening in addition to the blowoff, for washing out the boiler, the openings to be fitted with brass plugs.

Page 68, Par. 268. Change to read as follows: Threaded Openings. An opening in a boiler for a threaded pipe connection 1 in. in diameter or over shall have not less than the number of threads given in Table 7. If the thickness of the material in the boiler is not sufficient to give such number of threads, there shall be a pressed-steel flange, bronze-composition flange, steel-cast flange or steel plate, so as to give the required number of threads.

STAMPINGS ON SAFETY VALVES

Par. 269. Change to read as follows: Safety-Valve Requirements. Each boiler shall have two or more safety valves, except a boiler for which one safety valve 2 in. size or smaller is required by these Rules; in which case one or more may be used.

Page 69, Par. 273. Change to read as follows: Each safety valve shall be plainly marked by the manufacturer. The markings may be stamped on the body, cast on the body or stamped or cast on a plate or plates riveted to the body and shall contain the following:

a. The name or identifying trademark of the manufacturer.

b. The nominal diameter with the words "Bevel Seat" or "Flat Seat."

c. The steam pressure at which it is set to blow.

d. The lift in inches of the valve disk from its seat, measured at a pressure 3 per cent. higher than that at which the valve is set to blow.

e. The weight of steam discharged in pounds per hour at a pressure 3 per cent. higher than that for which valve is set to blow.

Pages 70, 71, 72, Table 8. Table 8 to be omitted.

Page 73, Par. 275a. Change to read as follows: Safety valve capacity may be checked in any one of three different ways, and if found sufficient, additional capacity need not be provided:

a. By making an accumulation test, that is, by shutting off all other steam discharge outlets from the boiler and forcing the fires to the maximum. The safety-valve equipment shall be sufficient to prevent an excess pressure beyond that specified in Par. 270.

Par. 279. Add the following: Where discharge pipes are used, ample drainage shall be provided at or near the safety valve.

Page 74, Par. 282. Change to read as follows: For purposes of inspection and to insure the valve being free, each safety valve used on a boiler shall have a substantial lifting device by which the valves may be raised by an amount equal to one-twentieth of the nominal diameter of the valve up to the maximum limit of $\frac{1}{16}$ in. when there is no pressure on the boiler.

Par. 283. Add the following: The seats and disks of safety valves shall be nonferrous material. The seat of a safety valve shall be fastened to the body of the valve in such a way that there is no possibility for the seat to lift.

Par. 284. Add the following: Springs used in safety valves shall not show a permanent set exceeding $\frac{1}{32}$ in. ten minutes after being released from a cold-compression test closing the spring solid. And the spring shall be so constructed that the valve can lift from its seat one-tenth the diameter of the seat before the coils are closed or before there is other interference.

Par. 286. Change to read as follows: All dimensions shall conform to the American Standard given in Tables 15 and 16 of the Appendix for the pressure therein specified, except that the face of the safety-valve flange and the nozzle to which it is attached shall be flat and without the raised face for pressures up to and including 250 lb. per sq.in. For higher pressure the raised face shall be used.

Page 75, Par. 291. Add the following: The lowest permissible water level for various classes of boilers is given in Par. 430 of the Appendix.

Par. 292. Change to read as follows: No water-glass connection shall be fitted with an automatic shutoff valve, except when the automatic shutoff valves are so constructed that the two connections to the water glass can be blown through separately and the steam connection cannot be entirely closed thereby.

Par. 296. Add the following: Where the use of a long pipe becomes necessary a shutoff valve or cock arranged so that it can be locked or sealed open may be used near the boiler. Such a pipe shall be of ample size and arranged so that it may be cleared by blowing out.

Page 76, Par. 299. Change to read as follows: Nozzles and Fttings. Flanged cast-iron pipe fittings used for boiler parts, for pressures up to and including 160 lb. per sq.in. shall conform to the American Standard given in Tables 15 and 16 of the Appendix, except that the face of the flange of a safety valve, as well as that of a safety-valve nozzle, shall be flat and without the raised face. For pressures above 160 lb. per sq.in., steel-cast and wrought-steel fittings shall be used for boiler parts with exceptions specified in Pars. 9 and 12. An allowable variation of 20 per cent. from the flange thickness required by Tables 15 and 16 may be made for steel-cast and forged-steel fittings, leaving the drilling of bolt holes unchanged. For pressures above 250 lb. per sq.in., the flange thickness and the thickness of the bodies shall be increased to keep within the same deflection limits and to give at least the same factor of safety as the fittings specified in Tables 15 and 16. The face of the flange of a safety valve, as well as that of a safety-valve nozzle, shall have a flat face for pressures up to and including 250 lb. per sq.in. and shall have raised face at higher pressures. Tables 15 and 16 do not apply to flanges on the boiler side of steam nozzles or to flanges left by the manufacturer as part of the boiler, and do not apply to fittings designed as part of the boiler.

Par. 307. Change to read as follows: Blowoff Piping. A surface blowoff pipe shall not exceed 1½-in. pipe size and shall form a continuous passage with the blowoff pipe external to the boiler. The internal and external pipes shall be separate with clearance between their ends and arranged so that the removal of either will not disturb the other. A brass or steel bushing, as shown in Fig. 19, or flanged connection shall be used.

Page 77, Par. 311. Change to read as follows: a. When the maximum allowable working pressure exceeds 125 lb. per sq.in., on all boilers except those used for traction and portable purposes, each bottom blowoff pipe shall have two valves, or a valve and a cock, and such valves, or valve and cock, shall be extra heavy, except that on a boiler having multiple blowoff pipes, a single master valve may be placed on the common blowoff pipe from the boiler, in which case only one valve on each individual blowoff is required.

PRESSURES ON PORTABLE BOILERS

b. On all traction and portable boilers when the maximum allowable working pressure exceeds 125 lb., per sq.in., each bottom blowoff pipe shall have one extra heavy valve.

Par. 315. Make the last sentence read as follows: The feed pipe shall be carried through the head or shell near the front end in the way specified for a surface blowoff in Par. 307 and be securely fastened inside the shell above the tubes.

Page 79, Par. 325. Change to read as follows: Lugs or brackets, when used to support boilers of all types shall be properly fitted to the surfaces to which they are attached. The shearing and crushing stresses on the rivets used for attaching the lugs or brackets shall not exceed 8 per cent. of the strength given in Pars. 15 and 16.

Par. 328. Change to read as follows: A water-tube boiler which is fired by hand shall have the firing door or doors of the inward opening type unless such doors are provided with substantial and effective latching devices to prevent them from being blown open by pressure on the furnace side.

Page 80, Fig. 20. Change Fig. 20, Par. 332, to read as follows:

(Name of State)
.....STD
Year put into service.
Safe Working Pressure.
(Name of Builder)
.....

Page 81, Par. 339. Change to read as follows: A boiler to be used exclusively for low-pressure steam heating may be constructed either of cast-iron, steel cast, or wrought iron or steel or any combination of these, but in all cases the connecting-rods and bolts shall be wrought iron or steel.

Page 82, Par. 343. Change to read as follows: In a hot-water boiler to be used exclusively for heating buildings or not-water supply, when the diameter does not exceed 60 in. and the grate area does not exceed 10 sq.ft., or equivalent, as defined in Pars. 359 and 360, longitudinal lap joints will be allowed.

When the grate area exceeds 10 sq.ft., or equivalent, as defined in Par. 360, and the diameter of the boiler does not exceed 60 in. longitudinal lap joints will be allowed, providing the maximum allowable working pressure does not exceed 50 lb. per sq.in.

Page 84, Table 9. Title of Table 9 to be changed to read as follows: Allowable Sizes of Safety Valves for Steam-Heating Boilers, of Water-Relief Valves for Water-Heating Boilers, and of Hot-Water Supply Boilers.

Page 85, Pars. 359 and 360. Insert the following center head:

GRATE AREA

Par. 359. Change to read as follows: Double-Grate Down-Draft Boilers. In boilers of this type the grate area shall be taken as 1¼ times the area of the lower grate.

Par. 361. Change to read as follows: Steam Gages. Each steam boiler shall have a steam gage connected to the steam space or to the water column or its steam connection, by means of a siphon or equivalent device of sufficient capacity to keep the gage tube filled with water and so arranged that the gage cannot be shut off from the boiler except by a cock placed near the gage and provided with a tee or lever handle arranged to be parallel with the pipe in which it is located when the cock is open. Connections to gages shall be of brass, copper or bronze composition. The dial of a steam gage for a steam heating boiler shall be graduated to not less than 30 lb.

Page 86, Par. 365. Change to read as follows: Damper Regulators. When a pressure damper regulator is used, it shall be connected to the steam space of the boiler.

Page 87, Par. 373. Change to read as follows: Hot-water boilers for a maximum allowable working pressure not exceeding 30 lb. per sq.in. used exclusively for heating buildings or for hot-water supply, when constructed of cast iron, steel cast, or wrought iron or plate steel or any combination of these, shall be subjected to a shop test of 60 lb. per sq.in. hydrostatic pressure applied to the boiler or the section thereof.

Par. 374. Change to read as follows: A maximum allowable working pressure in excess of 30 lb. per sq.in. will be allowed on a hot-water boiler constructed of cast iron, steel cast, or wrought iron or plate steel or any combination of these, used exclusively for heating buildings or for hot-water supply, provided such boilers or their sections have been subjected to a shop hydrostatic test of 2½ times the actual working pressure.

Par. 377. Change to read as follows: Name. All boilers referred to in this section shall be plainly and permanently marked with the manufacturer's name and the maximum allowable working pressure. All heating boilers built according to these rules may be marked A.S.M.E. standard.

Page 91, Par. 391a. Change to read as follows: Safety-valve capacity may be checked in any one of three different ways, and if found sufficient, additional capacity need not be provided;

a. By making an accumulation test; that is, by shutting off all other steam discharge outlets from the boiler and forcing the fires to the maximum. The safety-valve equipment shall be sufficient to prevent an excess pressure beyond that specified in Par. 270.

Page 93, Par. 402. Change to read as follows: When the maximum allowable working pressure exceeds 125 lb. per sq.in., the blowoff pipe shall be extra-heavy from boiler to valve or valves and shall run full size without reducers or bushings. All fittings between the boiler and valve shall be steel or extra-heavy fittings of bronze, brass, malleable iron or cast iron.

Page 107, Par. 421. Change to read as follows: Method of Computing Discharge Capacity. The required discharge capacity of a safely valve or valves for a boiler may be based either on the heat units in the fuel consumed or on the amount of steam generated.

The number of heat units that each safety valve will handle, for valves of the ordinary types in which the discharge capacity is proportioned to the lift, may be obtained as follows:

$U = 161{,}000 \times P \times D \times L$ for Bevel Seats at 45 deg.

$U = 227{,}500 \times P \times D \times L$ for Flat Seats

The amount of steam that a valve will discharge may be found as follows:

$W = 110 \times P \times D \times L$ for Bevel Seats at 45 deg.

$W = 155 \times P \times D \times L$ for Flat Seats

Where

U = Number of heat units per hour that a safety valve will handle, B.t.u.

W = Quality of steam that a safety valve will handle per hour, lb.

P = Absolute boiler pressure or gage pressure plus 14.7 lb. per sq.in.

D = Inside diameter of valve seat, in.

L = Vertical lift of valve disk, measured with 3 per cent. excess pressure, in.

Par. 422. Change to read as follows: The maximum quantity of fuel C that can be burned per hour at the time of maximum forcing is determined by a test. The maximum number of heat units per hour, or C × H is then determined, using the values of H given in Par. 427. The weight of steam generated per hour is found by the formula,

$$W = \frac{C \times H \times 0.75}{1100}$$

The sum of the safety-valve capacities marked on the valves shall be equal to or greater than W.

Page 108, Par. 423. Replace matter after statement of Example by:

$C \times H = 2150 \times 12{,}100 = 26{,}015{,}000$

$W = C \times H \times 0.75 \div 1100 = 17{,}740$

A 3½-in. bevel-seated valve with 0.11 in. lift would discharge in heat units

$U = 161{,}000 \times 239.7 \times 3\frac{1}{2} \times 0.11$

$= 14{,}858{,}000$

and in weight of steam

$W = 110 \times 239.7 \times 3\frac{1}{2} \times 0.11$

$= 10{,}150$

From which it can be seen that either method indicates that two such valves will give the proper relieving capacity.

Par. 424. Replace matter after statement of Example by:

$C \times H = 2000 \times 6400 = 12{,}800{,}000$

$W = C \times H \times 0.75 \div 1100 = 8730$

A bevel-seated 3½-in. valve is marked by the manufacturer 0.11 in. lift and discharge capacity for 100 lb. pressure = 4840 lb.; hence two such valves would be required.

Par. 425. Replace matter after statement of Example by:

$C \times H = 1000 \times 18{,}500 = 18{,}500{,}000$

$W = C \times H \times 0.75 \div 1100 = 12{,}620$

A bevel-seated 2½-in. valve is marked by the manufacturer 0.08 in. lift and discharge capacity for 275 lb. pressure = 6350 lb.; hence two such valves would be required.

Par. 426. Replace matter after statement of Example by:

$C \times H = 3000 \times 960 = 2{,}880{,}000$

and

$W = C \times H \times 0.75 \div 1100 = 1960$

A bevel-seated 2-in. valve is marked by the manufacturer 0.07 in. lift and discharge capacity for 150 lb. pressure = 2500 lb.; hence one such valve would be required.

Page 109, Par. 427. Change to make first sentence read as follows: For the purpose of checking the safety-valve capacity as described in Par. 422, the following values of heats of combustion of various fuels may be used.

Also change headings in tables which follow to make them read:

H = B.t.u. per lb.

and

H = B.t.u. per cu.ft.

Pages 113-114, Par. 430. Make first line to read as follows: Each boiler may have one or more fusible plugs located at the lowest permissible water level as follows:

Add at the bottom of p. 114 (Par. 430): v. Fire engine boilers are not usually supplied with fusible plugs. Unless special provision is made to keep the water above the firebox crown-sheet other than by the natural water level, the lowest permissible water level shall be at least 3 in. above the top of the firebox crown-sheet.

Cost Plus a Fair Profit

In its efforts to insure that the consumer gets anthracite coal at moderate prices, the Federal Trade Commission is requiring operators to report weekly all orders accepted, together with the prices. The producers of a very great proportion of the tonnage are selling their output at moderate prices, so that the high premiums charged by a number of operators during recent weeks are beginning to disappear from the market.

As a basis for publicity, within the powers of the commission, costs of production are being obtained from any operators who continue to maintain unjustifiably high prices. The commission is also getting monthly production statistics by sizes from all the anthracite mining companies, and anthracite jobbers are required to furnish the commission a weekly report, giving complete purchase and sales data, and gross profits on each transaction. With this information the commission can trace all jobbed anthracite from the mines to the retail yard.

Agents are in the field keeping in close touch with the prices charged by retailers in various parts of the country, so that distribution and prices are being traced all the way from the mine to the consumer.

Precautions in Fuel-Oil Storage*

By P. A. Colwell†

SYNOPSIS—Insurance records show that the number of fires in oil-storage plants is no greater than in plants using coal. The mains to large storage tanks are used to ground the tanks for protection against electrical storms. Underground tanks inclosed in concrete most desirable for storage in small plants. All oil should drain back into the tank. Tanks are provided with automatic fire extinguishers.

We will consider fuel oil only, not crude oil, which is sometimes used. The latter is much more hazardous from an insurance viewpoint, containing volatiles such as gasoline, and is an unrefined oil. Fuel oil, for burning purposes, should have a flash point of not less than 150 deg. F. That in common use locally averages 160-175 deg. F., and the burning point is about 220 deg. It is of such a dense nature that it is necessary to heat it by steam coils placed in the tank before it can be successfully pumped and burned. It is of an asphaltum base.

It may be of interest to first briefly describe the method of storage of fuel oil, taking a large local station as an example. Oil is stored in steel riveted tanks of large capacity. The tops are covered with tight steel roofs. Each tank is provided with a vent pipe which is screened at the outlet. Each tank is surrounded by an earthen and concrete embankment forming an inclosure of more than sufficient size to hold the entire contents of the tank in case of rupture thereof. The oil steamers tie up at the adjacent dock, and from them the oil is pumped through mains to the tanks. These mains, connecting with tanks and thence running under the earth, are relied upon to furnish grounding and thus protect the tanks during electrical storms. Each tank is also provided with an automatic fire extinguisher, a new and very efficient extinguisher for fires in volatile substances, the discharged fluid being of a soapy nature, spreading over the surface of the oil upon release by the automatic device, thus cutting off any air supply and preventing further combustion. Steam coils at the bottoms of the tanks keep oil at the proper temperatures for pumping. There is the necessary boiler house for furnishing this steam and the steam for pumping to tank cars on the adjoining spur track and to the tank trucks.

HIGH- AND LOW-PRESSURE SYSTEMS

There are two types of equipment in use—the low- and the high-pressure systems. The former is usually installed where steam pressures not exceeding ten or fifteen pounds are required and the latter where higher steam pressures are needed. The usual low-pressure burner used in this locality is of a rotary type, set in the furnace of the boiler, surrounded by a firebrick lining, the rapid rotation of the burner atomizing the oil, and a small electric motor pumps the oil from the supply tank to the burner. In high-pressure systems the oil is forced by a steam pump to the burners, where steam is also admitted, vaporizing the oil, the force of the steam driving the flame generated into the firebox, the burner being located outside of, or at the front of the furnace.

Where lot space permits, storage of oil is generally outside of the building in buried metal tanks. If there is no space around the building available for such storage tanks, as there usually is not around a down-town building, the tank is allowed to be placed under the basement floor or under the grade floor if there is no basement. Such a tank, when set under the building, must not exceed 9000 gal. capacity, according to the Underwriters' requirements. If the tank is outside of the building and buried within ten feet of the walls, 500 gal. is the limited capacity, unless such tank is incased in six inches of concrete, in which case the size allowed may be increased. The reason for restricting the size of the tank buried outside and near a building (when not protected by concrete) to 500 gal. capacity is that should this tank leak, it would be possible for the escaping oil to penetrate into the basement of the building, especially if the walls thereof were not tight, and this oil might accumulate in some dark or neglected spot and become a source of danger. Should the tank buried beneath the floor leak, its contents would probably drain off into the earth.

When the tank is buried beneath the floor, a reinforced-concrete covering is required. In many cases in the business district of this city, tide water interferes with excavating or it makes excavating expensive by requiring good waterproofing. In such cases the tanks have been allowed to be placed as deep as was practicable and the part then remaining above the floor level has been inclosed in 8-in. brick walls, or reinforced-concrete inclosure, and the space between the tank and its surrounding walls has been filled with earth and the top covered with a reinforced-concrete slab; the latter to prevent rupture should floors fall thereon.

TREATMENT OF STORAGE TANKS

The tanks vary in thickness from No. 14 U. S. gage metal for 500-gal. sizes to ¼ in. for sizes above 500 and up to 5000 gal., and they are tar coated on the outside for protection against rust. They are usually 3 ft. below the level of the basement or the ground, and they must be below all pipes running to the burners in boilers, so that all surplus oil will drain back to the tanks when the pumps have been shut down. Should the tank not have been burned deep enough to locate the top of it below the pump and burners, the suction pipe from tank to pump must enter the tank at the top, in which case the only oil not draining back would be that between the elbow in the suction pipe at the top of tank and the burners, and should be beneath the floor and protected from mechanical injury.

Vent pipes are required on all tanks. Small tanks may have a combined filling and vent pipe so arranged that the filling pipe cannot be opened without opening the vent pipe, and these pipes then terminate in a locked metal box. When a tank of a larger capacity than 1000 gal. is used, it must have a vent pipe at least 1 in. in size, extending at least 12 ft. above the ground, terminating in a screened gooseneck fitting. This fitting must not be near a window where escaping vapors could find entrance, or from which window fire might be communicated to the vapor by an open light. The screen would prevent a flare-back into the tank in case of ignition of vapor at the terminal of the vent pipe. Venting into boiler stacks has also recently become common practice.

All pipes for filling tanks which are located under floors of buildings and which run underground from the tanks to the filling caps outside must pitch toward the tanks.

Indicators are desirable for showing the depth of oil in tanks, but when used only those of a type which minimizes the exposure of the oil are allowed, and any device that would allow escape of oil when broken would not be approved.

Glass gages are not approved, but if absolutely necessary, substantial protection against breakage is required. No pet-cock can be used on any part of the oil-conveying system. This is to prevent any drip from being carelessly left open.

HANDLING THE LOW-PRESSURE SYSTEM

In the low-pressure systems a small electric motor forces the oil tanks into the rotary burners. In the hot-air pipe there is a thermostat control so designed that should the flame at the burner be extinguished while the motor is running, the cooling down would result in the shutting down of the motor within two minutes, the thermostat control throwing the rheostat to the first position. The height of the flame is regulated by a valve, and the pressure on the system is regulated by another valve placed on the return pipe to the tank from the pump, the closing down of which increases the pressure to the desired point. There is also an automatic control valve governing the oil supply to the burner, it being connected to the usual automatic damper control of the boiler by a rod, this device keeping the building at the desired temperature, the damper-control balanced rod being fitted with weights on one end.

In a high-pressure system a duplex steam pump is used to convey the oil from the tank to the burners. These pumps are fitted with relief valves so that should the pressure generated on the oil exceed the usual 25-30 lb., the surplus oil is discharged back into the tank through the return circulation pipe from the burners.

In case the tank has not been placed below the burners, but has been placed in an approved inclosure (as previously described) and the top of the tank is higher than the pump, the feed pipe from tank to pump is required to enter the tank at the top, thence extending to the bottom, to avoid gravity feed from tank to pump.

Shutoff valves are installed on feed and circulating pipes as near the tank as possible. They are also required on each

*Abstract of a paper read before the Providence Engineering Society, Providence, R. I.

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burner. In high-pressure systems a steam governor is installed at each burner to maintain even pressure. Piping from tanks to pumps and burners must be carefully installed, made tight, and must be exposed as little as possible to mechanical injury.

The burners should be large enough to furnish only sufficient oil for the maximum burning conditions, with control valve wide open. Burners containing oil-carrying parts or parts subject to intense heat should not be used, and burners should be of such design as to allow easy cleaning.

The preceding is an outline of the Underwriters' requirements. The following note prefaces the regulations in the pamphlet covering "Storage and Use of Fuel Oil":

"Apparatus using oil for fuel, however safeguarded, introduces a distinct increase in hazard, which should be recognized."

Does the fire record warrant this statement? I know of no fire in Providence having been caused by its use, and the fire commissioners' office also has no record of any. It will be of interest, in this connection, to turn to the records of the N. F. P. A., covering 17 years, up to and through 1912. They have no tabulated data since that date. Their figures show that its use is mildly hazardous, but since 1912 the methods of installation and use have certainly improved, as have the devices for handling and burning, so that it would not be fair to base our judgment entirely on these figures. The number of fires reported, caused by the use of fuel oil, 64, is certainly very small for a 17-year period.

Pittsburgh Smoke Ordinance

The following is the new smoke ordinance (No. 566) of Pittsburgh:

An Ordinance—Regulating the production or emission of smoke from any chimney, smoke-stack or other source within the corporate limits of the City of Pittsburgh, except buildings used exclusively for private-residence purposes and flats and apartment houses in which there are less than five apartments, and prescribing penalties for the violation of the provisions.

Section 1. Be it ordained and enacted by the City of Pittsburgh, in Council assembled, and it is hereby ordained and enacted by the authority of the same. That in the interpretation of this ordinance, unless the context otherwise requires:

"Persons" shall be considered as referring to all individuals, partnerships or associations other than corporations.

"Corporations" shall be considered as including all bodies corporate, joint stock companies or associated, domestic and foreign, their lessees, assignees, trustees, receivers and other successors in interest having any of the powers or privileges of corporations not possessed by individuals, partnerships or unincorporated associations.

"Chart" shall be considered as referring to the Ringelmann Smoke Chart as published and used by the United States Bureau of Mines.

"Stack" shall be defined as including chimneys, smoke-stacks, structures and openings of any kind whatsoever, capable of emitting smoke. Smoke-stacks on locomotive roundhouses shall be deemed parts of locomotives beneath them for the time being.

"Advisory Board" shall be considered as referring to the Board of Engineers appointed by the Mayor to act as advisors on engineering questions to the bureau chief, Bureau of Smoke Regulation of the city.

Section 2. The production or emission within the city, of smoke, the density or shade of which is equal to, or greater than No. 3 of the Ringelmann Chart, from any stack, except that of a locomotive or steamboat, for a period, or for periods aggregating two minutes or more in any period of 15 minutes, and the emission of such smoke from any locomotive or steamboat for a period or periods aggregating one minute or more in any period of eight minutes, except for a period not to exceed 20 consecutive minutes, not to exceed once a day, while a new fire is being built therein, is hereby prohibited.

Section 3. The provisions of this ordinance shall not apply to buildings used exclusively for private-residence purposes, except flats and apartment houses in which there are more than five apartments.

Section 4. No person or corporation shall construct, install, reconstruct, alter or repair any furnace, boiler-furnace, stack or other apparatus connected with stack, unless he or it shall make application in writing to the Bureau of Smoke Regulation on the form furnished by the said bureau, duly sworn to before a notary public or any person authorized to administer oaths, for a permit for such construction, installation, reconstruction, alteration or repair and in and by such application shall give the plans and specifications, showing the style and dimensions of the furnace, boiler-furnace, stack or other apparatus connected with a stack intended to be used, a description of the building or part thereof in which such furnace, boiler-furnace or other apparatus is located, including the means provided for regulating the temperature of such building or part thereof and ventilating the same, and generally all provisions made for preventing smoke, together with a statement of the kind of fuel proposed to be used and of the operating requirements to be made of the furnace or furnaces referred to therein and unless such application shall be passed upon by the Bureau of Smoke Regulation and approved in writing and a permit issued as hereinafter provided; provided, however, that minor or emergency repairs which do not increase the capacity of such furnace or which do not involve any substantial alteration in such furnace, boiler-furnace, stack or other apparatus, and which do not involve any alteration in the method or efficiency of smoke prevention, may be made without a permit.

Any application shall be approved or rejected within ten days after the same is filed in the office of the Bureau of Smoke Regulation.

PERMITS MUST BE OBTAINED

Upon the approval of any application a copy of which shall be left on file in the office of the Bureau of Smoke Regulation, and upon the payment of the fees hereinafter provided, the Bureau of Smoke Regulation shall issue a permit for the construction, installation, reconstruction, alteration or repair of such furnace, boiler-furnace, stack or other apparatus.

In the event that any such application is rejected by the bureau chief, Bureau of Smoke Regulation, the applicant has the right to appeal from his decision to the Advisory Board. Such appeal shall be made in writing to the bureau chief, Bureau of Smoke Regulation, who shall call a special meeting of the Advisory Board within three days for the consideration of the matter. If a majority of the members of the Advisory Board present shall be of the opinion that the application calls for such construction, installation, reconstruction, alteration or repair of furnace, boiler-furnace, stack or other apparatus, that there will not under reasonable conditions of operation be produced or emitted from the stack connected therewith, such smoke as is herein prohibited, the decision of the bureau chief, Bureau of Smoke Regulation, shall be reversed and the finding of the Advisory Board shall be binding upon the bureau chief, Bureau of Smoke Regulation; otherwise the same shall be confirmed. In which latter case the fees to the Advisory Board are to be paid by appellant, who shall first give bond or make other deposit of the funds for the amount of the fees provided for in Ordinance 161, Section 3, approved May 14, 1914.

Section 5. For examination of an application for a permit for any such construction, installation, reconstruction, alteration or repair, the Bureau of Smoke Regulation shall collect at the time of issuing such permit, a fee of $2.

The issue and delivery by the Bureau of Smoke Regulation, of any such permit, shall not be held to exempt the person or corporation to whom such a permit has been issued or delivered or who is in possession of the same, or whose application has been approved, from prosecution on account of the production or emission of smoke hereby prohibited.

Section 6. The bureau chief, Bureau of Smoke Regulation, shall keep in the office of the Bureau of Smoke Regulation all applications made, and a complete record thereof as well as of all permits issued. He shall also keep a record of all smoke observations of all stacks and generally of the work done by the Bureau of Smoke Regulation. All such records shall be open for inspection by the public at all reasonable times. He shall in all matters pertaining to the enforcements of the provisions of his ordinance report direct to the Director of the Department of Public Health.

RIGHT OF ENTRY FOR SMOKE INSPECTOR

Section 7. The bureau chief, Bureau of Smoke Regulation, assistant bureau chief, Bureau of Smoke Regulation, and the smoke inspectors shall have the right to enter in the performance of their duties at all reasonable hours, all premises from which smoke is being emitted or has been emitted, and any person who shall, after proper identification, deny admittance to such person or persons or interfere with him or them in the performance of his or their duties shall be liable to a fine not exceeding $100 or undergo an imprisonment in the Allegheny County jail or workhouse of not more than 30 days, or both, at the discretion of the committing magistrate or alderman.

Section 8. If any person or corporation shall violate any one or more of the prohibitions or requirements of this ordinance, he or it shall, upon conviction before any police magistrate or alderman of the City of Pittsburgh, be subject to a fine or penalty not exceeding $100 for each and every violation thereof, and each day's violation shall constitute a separate offense; or undergo an imprisonment in the Allegheny County jail or workhouse of not more than 30 days, or both, at the discretion of the committing magistrate or alderman.

Section 9. That any ordinance or part of ordinance conflicting with the provisions of this ordinance be and the same is hereby repealed so far as the same affects this ordinance.

Heating Convention at Detroit

On June 12 to 15 the National District Heating Association held its ninth annual convention in Detroit, headquarters being at the Hotel Ponchartrain. Although a number of the members usually active in convention affairs were missing, the attendance, for war times, was good and much interesting information was brought out in the papers and the discussions. In the way of entertainment Detroit lived up to its reputation, sight-seeing trips and amusements being arranged for the ladies and other entertainment for all attending. The sessions were short and snappy and the exhibits the best the association ever had.

Mayor Oscar B. Marx welcomed the conventioners to the city, George W. Martin responding for the association. The report of the executive committee included the list of new members, many of whom were from Toledo, where, through the efforts of A. C. Rogers, the first company section of the association had been formed. In his address President Gifford highly commended the various reports, the success of the association being due largely to the good work of the standing committees. He was in favor of more frequent publication of the "Journal" and urged the membership to make greater use of it. The president believed that advancement in central heating was to come along high-pressure lines, the advantages of exhaust steam having been overestimated greatly in the past. A committee to start work in this field would be advisable. In the combination plant it was important to know the cost of each service. Insuring a profit necessitated investigation of operating conditions, fixed charges and rates, the last-named being too low and especially under present conditions.

Secretary Gaskill's report showed practically an even balance between receipts and expenditures and up to June 1 a gain of 31 members over the figures given at the last convention, the total now being 383. Other executive business was the passage of a number of amendments to the constitution, all of which were published in the "Journal" of Jan. 15, 1917.

MORE ATTENTION TO FUNDAMENTAL LAWS

At the afternoon session, Tuesday, Prof. John R. Allen made a strong plea to heating engineers to pay more attention to the fundamental laws of heat. There had been too much experimental work on details with little attention given to working up practical data to introduce into fundamental laws. A great deal of information on constants for different materials was lacking. In working up these data there was great opportunity for coöperation between the engineer and the physicist. It was time to get away from rule-of-thumb methods. To illustrate, Professor Allen cited the laws governing convection and radiation, the transparency of building materials to heat rays and the storage of heat in the building and in the radiator itself. The discussion turned to heat storage and the advisability of shutting off or maintaining heat overnight. Except in the coldest weather leaving steam on overnight did not show a decided advantage. In reply to a question as to the best location of radiators, Professor Allen cited experiments being conducted at the University of Michigan confirming the old belief that in general the most economical place was under the windows.

The report of the station-operating committee dealt with feed-water treatment, specially designed furnaces and insulation of boiler walls, bonus systems, methods of determining the best ratio of heating and electric loads in connection with district heating and electric systems, and recent progress in station development. The first and the fourth of these subjects were given the most attention in the report. Discussion centered on feed-water treatment and insulation of boiler walls. To avoid cracking, the Detroit Edison Co. was using furnace walls of solid firebrick in order to have the same coefficient of expansion all the way through. Division of the wall into three parts was also being given a trial.

A. C. Rogers reviewed the educational work being done in the new company section at Toledo, the station-operating class having been especially successful. Secretary Gaskill had high praise for the efforts of the section. It was one way to help the men farther down, whom the national association did not reach. In the Ohio Electric Light Association the formation of station-operating, meter and other committees from the men actually engaged in the work had brought excellent results. Practical questions were discussed freely and with intelligence. The men were 25 per cent. better since they had been sent to these committee meetings.

At the first session Wednesday an interesting paper prepared by Nelson S. Thompson gave the operating characteristics of certain large Federal buildings in Washington served by the central heating, lighting and power plant, and of particular interest, methods of predicting the operating characteristics of buildings under construction. Empirical figures on steam for heating, lighting, power and elevator requirements, cost of repairs, supplies and labor were included. These data will be presented in a later issue of "Power."

Prof. J. D. Hoffman spoke on climatic conditions in Nebraska and the problems heating engineers had to meet in that state.

The session ended with the report of the Underground Construction Committee. Efforts had been made to analyze all methods of waterproofing submitted by members and otherwise, and of these the most effective and practical were submitted. It was the committee's opinion that "there was no such thing as a perfectly waterproofed steam or hot-water main." In the discussion it was claimed that concrete could be made water-tight.

In its report the Public Policy Committee discussed the following four topics: Economy among workmen, employer's liability and workmen's compensation laws, economy in the use of heating service and rates for heating service. Under the subject last named it was pointed out that steam heating could no longer be regarded as a byproduct of electric light and power and charged for at a low rate. The requirements of the two services do not coincide, and it has been the experience of the central-station companies in Detroit that the cost is about the same as in those stations where live steam only is produced for heating purposes. It was believed that the advantages and economies of the live-steam plant with high-pressure distribution and freedom from complications with the electric system more than offset the value of any additional electrical energy obtainable from turbo-generators in these plants. Besides, the cost of transmitting low-pressure steam is greater than that of transmitting steam at higher pressures, owing to the larger pipes required, which as a rule will more than offset any possible saving by the use of the exhaust steam. Accordingly, the committee recommended that public-utility companies, in figuring the cost of steam, eliminate all confusing data and figure simply on the cost of producing live steam from an independent plant.

Data on the adaption of cooking apparatus for low pressures and the steam consumption of various steam-using equipment for kitchens and laundries were presented in the report of the Educational Committee. There were also sections on surface-combustion heaters and on the adaption of sprinkler systems for use in building heating.

BUILDING REQUIREMENTS PER UNIT FLOOR SPACE AND VOLUME

The Heating Research Committee, working jointly with a committee from the National Electric Light Association, presented a most interesting report giving data on the steam and electrical requirements and the coal per unit of floor space and per unit of volume in various types of building. Policies to be pursued by the central-station companies in advising, supervising or operating the customer's steam plant in the event of his adoption of central-station service were outlined and the best means for carrying out these policies suggested.

Although the report of the Accounting Committee was not available at the meeting, an interesting discussion on the subject took place. President Gifford stated that the distribution of values and costs should be based upon the theory of combining the two departments of heating and electricity on the share-alike basis, as each department received its share of the benefit derived from the combination. The physical property should be distributed according to the demand made upon it by either service and the operating costs according to the use made of the apparatus. For example, take a condensing electric plant requiring 600 hp. in boilers and a heating plant requiring a maximum of 800 hp. when operating alone. The two might be taken care of by 800 hp. in boilers with a proper-sized unit in reserve. In this case it would be equitable to divide the boiler costs $^{6}/_{14}$ to electric and $^{8}/_{14}$ to heating. The same division should be made on the coal-storage and the coal- and ash-handling facilities. Real estate may be divided according to the use of it made by each department and the building cost according to the cubic contents occupied.

DIVISION OF EQUIPMENT COST

In the average combination plant prime movers are operating noncondensing the greater part of the heating season, thus reducing the capacity of the prime movers, and if this reduces the electric capacity, whatever this percentage of reduction represents in first cost less the value to the electric department during the nonheating season of having the larger apparatus is chargeable to the heating department. Ordinarily, the larger plants operate six months condensing and six months noncondensing. In this case 50 per cent. of the reduction is chargeable to electric and 50 per cent. to heating. For example, a unit of 5000 kw. at 80 per cent. power factor operating condensing may develop 4000 kw. at 80 per cent. power factor when operating noncondensing. Thus 10 per cent. (50 per cent. of 20 per cent.) of the first cost is chargeable to heating investment. The cost of piping to the prime mover is divided in the same way as the cost of the machine it serves.

The division of the operating costs should be made according to the use made of the apparatus and the judgment of the plant operator.

Thursday afternoon was devoted to a general discussion of rates, much of it being considered more or less confidential and

withheld from the proceedings. It was generally recognized that heating rates had always been too low and that the present cost of coal made higher rates imperative. To save the heating company, prompt revision was necessary. To secure prompt action, coal riders to contracts and retroactive agreements with state commissions were discussed and the coal situation in general rehearsed. Mention was made of the recent ruling of the Ohio State Commission against discrimination, which eventually would wipe out all contracts in relation to service. The same action was anticipated from all state commissions in the country. All customers must be billed at the same rate, so that a change in rates would immediately void a running contract.

Rather than follow the rates charged in other towns, the first essential in rate making was to determine costs. Tangible and intangible values of the property should be included in determining the fixed charges, and the operating charges were a matter of arithmetic. The tendency in these days was to allow a reasonable profit and it was not infrequent for state commissions to raise rates on their own initiative.

The report of the Meter Committee Friday morning discussed recent improvements in steam-flow meters and described some new condensation meters. A series of tests was outlined, which the committee recommended be considered as standard for determining the characteristics of the various types of condensation meter. Comparative operating data as to the number of troubles occurring with different types of meter were also given.

A discussion session followed in which three subjects were briefly considered: Loss of Heat Through Insulation, Ash and Coal Handling Problems and Efficiency of Consumer's Installations in Relation to Central Station Supply. A short paper by R. C. March giving data on a forced hot-water heating system introduced the first topic. In the second G. W. Martin referred to the increasing use of steam-jet systems for handling ash and in connection with the scarcity of coal referred to the burning of culm and coke breeze, neither of these fuels giving good results. Crushed coke mixed with bituminous coal or burned straight had been used with success, but in the latter case conditions had to be just right. G. J. Soucy had burned culm with success years ago, the fuel being hand-fired and perhaps of better quality than present-day culm.

The third topic, the efficiency of consumers' installations, was introduced in a paper by G. L. Flechsenhaar. The author believed that proper treatment of the consumer was one of the most valuable assets in the heating business and that coöperation must exist in all cases to further the interests of both the company and the consumer. Thorough inspection of the piping and radiation in the consumers' premises, design of heating system, covering of steam pipes, the keeping of records to show a complete summary of the operation of each installation, were all factors of importance to which attention might be given with profit.

At the end of the session a number of resolutions were passed, one of them requesting the Government to regulate and control the price of coal and in transportation give it primary right of way. Another offered the services of the association to the Government. Officers elected during this session were: G. W. Martin, president; J. C. Hobbs, first vice president; J. L. Hecht, second vice president; O. M. Rau, third vice president; D. L. Gaskill, secretary-treasurer; J. H. Walker and A. C. Rogers, executive committee.

Friday afternoon and evening were given up to a boat ride to the St. Clair Flats. Other entertainments were a most enjoyable banquet Wednesday evening and on the following night a theater party. Throughout the four days card and tea parties and automobile sight-seeing trips were arranged for the ladies.

New York State N. A. S. E. Convention

The twenty-second annual convention of the New York State Association of the N. A. S. E. was held at Rochester, June 14, 15 and 16. The Seneca Hotel was the headquarters. The main ballroom on the second floor was placed at the disposal of the convention committee. One end of the hall was screened off for the sessions of the delegates, while around the other and larger end tables were arranged for the use of the exhibitors for the display of light material, catalogs and other literature. There were upward of twenty-five exhibitors and a list of sixty patrons.

While the attendance of delegates was not fully up to the standard, the interest manifested in the several important measures brought up was gratifying to those present and the meetings proved very interesting and instructive. The principal item of discussion was the necessity of a concentrated effort on the part of all the operating engineers' associations of the State of New York to agree upon a uniform license bill. The convention went on record for a state license law rather than laws for municipalities as now exists.

On Thursday evening there was a reception to the delegates and guests at the Engineers' Hall, where music and a general good time was had.

The official opening of the convention took place in the Rose Room of the Seneca Hotel on Friday morning, George C. Van Vechten, chairman of the local committee, presiding. C. A. Poole, city engineer, welcomed the convention to Rochester. William Downey, state president, responded and thanked Mr. Poole on behalf of the engineers. F. Harper Sibley, president of the Chamber of Commerce, was glad that Rochester had been selected for the meetings; he spoke of the beauties and conveniences of the city and hoped that the next national convention would come there. William J. Reynolds, past national president, representing the national president of the N. A. S. E., fittingly responded to Mr. Sibley. George B. Schoeffel, of Rochester, also welcomed the convention to the Garden City and was responded to by William Aydelotte, past state president. William Bedod gave a five minutes' talk on the "National Engineer." William Downey then appointed the necessary committees, and the meeting adjourned until the afternoon session at 2 o'clock.

On Friday evening Harry Ashton, of the Ashton Valve Co., gave an instructive lecture on the safety valve. F. Grey Maxwell, by courtesy of the Metric Packing Co., played several selections on the violin. Jack Armour, of "Power," told stories and recited.

On Saturday afternoon the convention enjoyed an outing at Seneca Park. Baseball and outdoor sports of every kind were indulged in. There were prizes for the winner in each event. The ladies of the auxiliary furnished luncheon. There was a theater party for the ladies on Friday evening and an auto ride on Saturday morning.

At the closing session of the delegates on Saturday morning the following officers were elected: George C. Van Vechten, president, Rochester No. 3; Patrick F. Cassidy, vice president, Phoenix No. 24; William Roberts, secretary, Yonkers No. 28; William Downes, treasurer, Manhattan No. 25; George Ely, conductor, Rochester No. 3; Fred J. Desmond, doorkeeper, Rochester No. 14; Andrew S. Bennett, chaplain, Brooklyn, No. 41; J. B. McGowan, state deputy, Brooklyn No. 8. William J. Reynolds, past national president, installed the officers. Retiring state president William Downey was presented a handsome mahogany clock and silver bonbon tray, the gift of the delegates. Brooklyn was chosen for the next annual meeting in June, 1918. Memorial services, conducted by the Rev. George H. Fickes, of the Dewey Avenue Presbyterian Church, closed the meeting.

PERSONALS

Frank Beers, formerly of the Green Economizer Co., has recently become associated with the Roto Co., of Hartford, Conn.

C. A. Newman, formerly manager of sales promotion for Henion & Hubbell, Chicago, wholesalers in power pumps, mining and mill supplies, has been made sales manager of the Boiler-Kote Co., with general sales offices in the Fisher Building, Chicago.

ENGINEERING AFFAIRS

The Michigan State Association, N. A. S. E., will hold its convention at Detroit, June 27 to 29.

The New England States Association, N. A. S. E., will hold its convention at Portland, Me., on June 28 to 30.

The National Committee on Gas and Electric Service was organized some months ago for the purpose of securing to the United States Government the most effective coöperation in the conduct of the war from the gas and electric-light and power industries. The membership of this committee is composed of representative men from the National Electric Light Association, the American Gas Institute and the related natural gas industry. John W. Lieb, vice president of the New York Edison Co., is chairman of the organization. The committee is recognized by the Advisory Commission of the Council of National Defense as one of its advisory committees and has established official headquarters in the Munsey Building, Washington, D. C., where it has placed itself at the service of the Government for any assistance it may be able to render in behalf of the public-utility industries it represents, and is prepared to act as a vehicle of communication between the various instrumentalities organized by the Government for the conduct of the war and the gas companies producing and distributing artificial and natural gas and the light and power companies furnishing electric service, in all matters relating to national defense

THE COAL MARKET

Boston—Current quotations per gross ton delivered alongside Boston points as compared with a year ago are as follows:

ANTHRACITE

	Circular[1]		Individual[1]	
	June 23, 1917	One Year Ago	June 23, 1917	One Year Ago
Buckwheat ..	.....	$2.05—3.20	$6.00—6.75	$3.25—3.50
Rice	.....	2.50—2.65	5.20—5.85	2.70—2.95
Boiler	.....	.....	..... *	.....
Barley	.....	2.20—2.35	4.90—5.00	2.35—2.60

*No prices quoted.

BITUMINOUS

Prices per gross ton for Boston delivery are as follows:

	F.o.b. Mines*		Alongside Boston†	
	June 23, 1917	One Year Ago	June 23, 1917	One Year Ago
Clearfields ..	$5.25—6.00	$1.00—1.55	$8.75—10.00	$4.25—5.00
Cambrias and Somersets..	5.50—6.25	1.30—1.75	9.25—10.00	4.60—5.40

Pocahontas and New River, f.o.b. Hampton Roads, is $7.25—7.50, as compared with $2.75—2.85 a year ago; on cars Boston price is $11.25—12.

*All-rail rate to Boston is $2.60. †Water coal.

New York—Current quotations per gross ton f.o.b. Tidewater at the lower ports* as compared with a year ago are as follows:

ANTHRACITE

	Circular[1]		Individual[1]	
	June 23, 1917	One Year Ago	June 23, 1917	One Year Ago
Buckwheat ..	$4.00—4.15	$2.75	$5.00—5.25	$2.45—2.75
Rice	3.40—4.05	2.25	4.00—4.25	2.20—2.25
Barley	2.90—3.10	1.75	3.00—3.25	1.70—2.00

BITUMINOUS

	South Amboy	Port Reading	Mine Price
Clearfield	$6.75—7.00	$6.75—7.00	$5.25—5.50
South Forks	7.00—7.25	.....	5.50—5.75
Nanty Glo	7.00—7.25	.....	5.50—5.75
Somerset	6.75—7.00	6.75—7.00	5.25—5.50
Quemahoning	7.00—7.25	7.00—7.25	5.50—5.75

*The lower ports are: Elizabethport, Port Johnson, Port Reading, Perth Amboy and South Amboy. The upper ports are: Port Liberty, Hoboken, Weehawken, Edgewater or Cliffside and Guttenberg. St. George is in between and sometimes a special boat rate is made. Some bituminous is shopped from Port Liberty. The freight rate to the upper ports is 5c. higher than to the lower ports.

Philadelphia—Prices per gross ton r.o.b. cars at mines for line shipment and f.o.b. Port Richmond for tide shipment are as follows:

	Line		Tide	
	June 23, 1917	One Year Ago	June 23, 1917	One Year Ago
Buckwheat ...	$2.90	$1.65	$3.80	$2.55
Rice	2.40	1.00	3.40	....
Boiler	2.20	.90	3.30	1.80
Barley	1.90	.75	2.15	....

Pittsburgh—Price of steam coal per net ton f.o.b. mine, Pittsburgh district:

	June 23, 1917	One Year Ago
Slack	$5.00	$1.65—1.90
Mine-run	4.75—5.00	1.75—2.00
¾-in.	4.75—5.00	1.85—2.10

Add 40c. per ton for freight charge to Pittsburgh.

Chicago—Current prices per net ton f.o.b. mines are as follows:

	Williamson and Franklin Counties	Saline and Harrisburg	West Virginia Smokeless	Springfield	Clinton and Sullivan Counties
Steam lump ..	$3.00—3.75	$3.00—3.50	$5.00—5.25	$3.00	$3.00—3.25
Lump	3.50—3.75	3.50—3.75	5.00—5.50	3.25	3.50—3.75
Egg	3.50—3.75	3.50—3.75	5.00—5.50	3.25	3.50—3.75
Nut	3.50—3.75	3.50—3.75	.....	3.25	3.50—3.75
No. 1 nut....	3.50—3.75	3.50—3.75	.....	3.25	3.50—3.75
No. 2 nut....	3.50—3.75	3.50—3.75	.....	3.25	3.25
No. 3 nut....	3.25—3.75	3.25—3.75	.....	.....	.....
No. 1 washed.	3.50—3.75	3.50—3.75	.....	.....	.....
No. 2 washed.	3.25—3.75	3.50—3.75	.....	.....	.....
Mine-run	2.75—3.50	2.75—3.50	5.00—5.25	3.00—3.25	2.50—3.00
Screenings ...	2.75—3.25	2.75—3.25	.....	2.25—2.75	2.25—3.00

Hocking lump, $4.50—4.75; splint lump, $4.25—4.75.

St. Louis—Prices per net ton f.o.b. mine a year ago as compared with today are as follows:

	Williamson and Franklin Counties		Mt. Olive and Staunton		Standard	
	June 23, 1917	One Year Ago	June 23, 1917	One Year Ago	June 23, 1917	One Year Ago
6-in. lump..	$3.50	$1.50	$2.25	$1.25	$2.25	$0.95
2-in. lump..	3.50	1.40	2.25	1.15	2.10	.85
Steam egg..	3.50	1.45	2.25	1.15	2.10	.85
Mine-run ...	3.00	1.10	2.25	1.00	2.00	.85
No. 1 nut...	3.50	1.50	2.25	1.45	2.25	1.25
2-in. screen..	2.60	.90	2.25	.85	1.75	.85
No. 5 washed	2.50	.70	2.25	.75	1.75	.75

Williamson-Franklin rate St. Louis, 72½c.; other rates, 57½c.

Birmingham—Current prices per net ton f.o.b. mines are as follows:

	Mine-Run		Mine-Run
Pratt	$3.75	Carbon Hill	$3.25—3.50
Big Seam	2.75—3.25	Cahaba	4.00
Black Creek	4.00		

[1]Individual prices are the company circulars at which coal is sold to regular customers irrespective of market conditions. Circular prices are generally the same at the same periods of the year and are fixed according to a regular schedule.

PROPOSED CONSTRUCTION

Ariz., Mesa—City will sell $125,000 bonds to construct an electric-light plant. W. A. Macdonald, Clk.

Idaho, Wallace—The Shoshone County Power Co., organized to build and operate a power transmission line to the Murray Dist. to supply power to the Yukon Gold Co. D. L. Huntington, H. L. Bleerer, C. S. McCalla and A. F. Steel, all of Spokane, are some of the incorporators.

Ind., Garrett—(Official)—The Baltimore & Ohio R.R. (Chicago Division) will construct a new power plant to provide power, heat and light for the terminals. The power plant will be 26 ft. high, 80 ft. long and 71 ft. wide, brick, being divided into an engine room and boiler room, also an elevated coal trestle for dumping coal. The company will move 2 of the present 250 hp. Aultman and Taylor horizontal water tube boilers from the old plant and will reset with a higher setting so as to accommodate the new stokers; 2 new Stirling type boilers of about the same capacity as the aforementioned boilers will be purchased and installed and Taylor stokers will be applied to all the 4 boilers. Natural draft for the boilers will be furnished by a brick stack about 7 ft. in diameter at the top and 160 ft. high. This natural draft will be supplemented by steam engine driven forced draft, equipment to be installed in connection with the stoker installation; 1 generator will be installed, about 225 or 250 k.v.-a. capacity and will be designed to deliver 440-volt 3-phase 60-cycle current. Company may move both 100 k.w. engine generators and an air compressor from the old plant and also install an additional air compressor, about 1350 cu.ft. of free air per minute capacity. Work may be done by the company's forces under the direction of J. H. Davis, Baltimore, Elect. Engr.

Iowa, Hamilton—The Keokuk Electric Co. plans electric transmission line from Hamilton to Warsaw.

Kan., Manhattan—The Rocky Ford Milling and Power Co. is extending its transmission line from Manhattan to Alma, to furnish electricity there.

Ky., Whitesburg—W. C. Daniel & Son plan to install an electric-light plant.

Mass., Southbridge—The Hamilton Woolen Co. will construct a 35 x 49-ft. boiler house. Estimated cost $10,000.

Mich., St. John—The Bd. of Public Works is considering enlarging the electric-light plant. The machinery to be installed will cost from $30,000 to $40,000.

Mo., Bernie—G. N. Laswell and C. L. Coble, Kennett, have secured a franchise to install an electric-light plant here.

Mo., Harrisonville—City plans to hold election to vote on $25,000 bonds to install an electric-light plant.

Mo., Kansas City—City plans to install an electric-light plant. Estimated cost $800,000. C. Hill, City Engr.

Neb., Bayard—City voted in favor of issuing $4500 bonds to install an electric-lighting system.

N. J., Bayonne—The Bayonne Steel Casting Co., Oak St., is having plans prepared for a 1-story, 30 x 110-ft. power house. Estimated cost $8000.

N. Y., Gardenville—The New York Central R.R. has had plans prepared for a power station and transfer station. C. L. Kittredge, Ch. Engr.

N. Y., New York—(Borough of Richmond)—The L. A. Dreyfus Co., Maple Ave., Rosebank, has had plans prepared for a 2-story power station. Estimated cost $25,000.

N. Y., New York—(Borough of Queens)—The Imperial Machine Co., 108 West 34th St., Manhattan, will construct a power house in Lindenhurst. Estimated cost $8000.

N. Y., Utica—The Globe Mills and Woolen Co., 819 Court St., is having plans prepared for a power plant.

N. Y., Yonkers—The National Sugar Refining Co. has had plans prepared for a 1-story, 80 x 193-ft. boiler house. Estimated cost $40,000.

N. D., Omemee—City is considering installing an electric-lighting plant.

Okla., Haileyville—City plans electric-light plant.

Ont., Hanover—City plans a hydro-electric system. W. Beamish, Engr.

Ont., Iroquois Falls—Abitibi Power and Paper Co. is having plans prepared for a steel high-tension tower transmission line.

Ont., Toronto—The Hydro-Electric Power Comn. of Ontario plans to extend its plant on University Ave. F. A. Gaby, Continental Life Bldg., Ch. Engr.

Penn., Allentown—The Lehigh Valley Power Co. plans an addition to its boiler house. Ford, Bacon & Davis, Engr., 115 Broadway, New York City, soon receives the bids.

Penn., Benton—The Benton Hydro-electric Co. has made application to the Public Service Comn. for permission to issue $75,000 bonds to extend its system. G. Appleman, Supt.

Penn., Ortanna—The Ortanna Electric Light and Power Co. has applied to the Public Service Comn., Harrisburg, for permission to issue $11,000 stock, to improve its plant.

Penn., Pittsburgh—The Duquesne Light Co. has had plans prepared for improving its plant at Broad and Beatty St., East End. Estimated cost $17,000. F. Uhlinhaut, Jr., 435 6th Ave., Ch. Engr.

Penn., Williamstown—The Lykens Valley Light and Power Co. has applied to the Public Service Comn., Harrisburg, for permission to issue $12,000 bonds to improve its plant.

Wis., Milwaukee—The Cooper Underwear Co. is having plans prepared by T. S. Watson, engr., Majestic Bldg., for a 50 x 75-ft. brick and steel power house and boiler room.

Wis., Milwaukee—Pawling & Harnischfeger will construct a brick and steel power house in connection with its plant at 38th and National Ave. Estimated cost, $30,000.